NEW ENGLAND WATER WORKS ASSOCIATION.

ORGANIZED 1882.

Vol. XIII. March, 1899. No. 3.

This Association, as a body, is not responsible for the statements or opinions of any of its members.

IMPROVEMENT IN PUMPING ENGINES.

BY. F. F. FORBES, SUPERINTENDENT, BROOKLINE, MASS.

[*Read Dec. 21, 1898.*]

Where real progress has been made in any direction, it is very interesting, and a cause for much satisfaction, to look back and review the struggles which have at last been successful, and also to fully realize that the hours and years of toil have not been in vain. Of the many parts which make up a water works system no one is more important than the pumping engine. In the first place the machine must be reliable, and it also should be accessible in all its parts, so that repairs can be quickly and easily made.

Economy in coal consumption is also a very important feature, as the demand for fuel is a continuous one, and during the last 20 years great gains have been made in reducing the amount of fuel required to pump a certain amount of water. As this economy has been accomplished mostly by the use of machines carrying the expansion of steam over a much greater range, I wish to trace in the briefest manner possible the progress made in this direction.

Twenty years ago the contract requirements for pumping engines were from 50,000,000 to 75,000,000 foot pounds for each 100 pounds of coal burned. Today, as by the specifications for the Chestnut Hill engine, the contract requires a duty of 150,000,000 foot pounds for each 1,000 pounds of dry steam. It is with some amusement that we now look back only a few years to those duty trials conducted with so much care and gravity, and think how much satisfaction the announcement of a duty of 60,000,000 or 70,000,000 gave to all those interested, and how the account of these trials appeared in the technical papers as great achievements

in pumping engines. Changes have rapidly taken place since those times, and we will hastily glance at some of them.

In 1882, sixteen years ago, the first Gaskell high duty crank and fly wheel engine was installed at Saratoga Springs, N. Y., and gave a duty of 112,899,980 foot pounds for each 100 pounds coal burned at the official test. Henry R. Worthington built his first high duty engine in 1885, which was sold to the city of New Bedford, Mass. This engine gave a duty on a 12-hour test of 79,238,160 foot pounds for each 100 pounds of coal burned. Now we come to a new departure in pumping machinery, the introduction of the triple expansion pumping engine. The credit for this new departure belongs to the Edw. P. Allis Co., of Milwaukee, Wis., and the first engine was made in 1886, only twelve years ago, for the city of Milwaukee. This engine had a capacity of 6,000,000 gallons. In 1888 the Holly M'f'g. Co. produced their first triple expansion engine. This engine was sold to Fort Wayne, Ind. In 1889 Henry R. Worthington installed his first triple expansion engine—one of 3,000,000 gallons capacity at Port Perry, Pa.

The Geo. F. Blake M'f'g. Co. made their first engine of this kind in 1890, one of 5,000,000 gallons capacity, and since that date all the other large builders of pumping engines have made more or less triple expansion engines. It may be interesting to note the aggregate daily capacity of triple expansion pumping engines in the United States for each year since they were made to 1898, and also the total daily capacity to the same date.

TABLE SHOWING GROWTH IN THE USE OF TRIPLE EXPANSION PUMPING ENGINES IN THE UNITED STATES.

Year.	Capacity of Triple Expansion Engines Made During the Year in Gallons.
1886	6,000,000
1888	123,500,000
1889	29,000,000
1890	29,000,000
1891	37,000,000
1892	96,250,000
1893	97,000,000
1894	60,500,000
1895	116,000,000
1896	187,375,000
1897	435,750,000
Total daily capacity	1,217,375,000 gallons.

These figures give us an idea of the marvelous growth in the use of engines of this class—a growth made imperative by the great saving they produce. Now what economical results may we confidently expect of the engine of today? We can safely answer that a triple expansion engine can be obtained at a moderate price, which will give a duty of 140,000,000 to 160,000,000 foot pounds for each 1,000 pounds steam used. Now think what this means. It means that today we can pump the same amount of water with $\frac{1}{3}$ to $\frac{1}{4}$ of the coal required 20 years ago, and although the money value of the coal saved in this way may be greater than the net saving owing to the larger capital invested, and the additional help which may be required in some cases, yet when the amount of water pumped per day is not larger even than 1,000,000 gallons, unless the head be very low, a high duty engine will be found much cheaper in the end.

An account of work done is always more convincing than theory, and in closing this brief history the following table is appended to show what the town of Brookline has saved in fuel by the use of a triple expansion, crank and fly wheel engine. This table includes all the coal used at this station for all purposes with no deduction for ashes or moisture. You will notice that with an engine of a capacity of 5,000,000 gallons per day, the average amount pumped was little more than $\frac{1}{4}$ of this quantity. Had the engine been in use more hours each day the economic results would have been much greater.

RECORD OF WORK DONE AT THE LOW SERVICE PUMPING STATION OF BROOKLINE, MASS.

Year.	Gallons Pumped.	Coal consumed in pounds for all purposes. No deductions.	Gallons per pound of coal.	
1890....	322,767,796	1,141,175	282.8	
1891...	356,812,572	1,263,395	282.4	
1892...	390,593,090	1,351,032	289.1	
1893 ..	443,973,099	1,582,471	280 5	
1894....	481,633,366	998,107	482.5	Triple expansion engine, in use part of the time.
1895....	483,969,131	803,642	602.2	In use most of the time.
1896....	494,095,562	745,692	662.6	In use all the time.
1897....	508,174,048	761,826	670.0	In use all the time.

Average number of gallons pumped per day in 1897.....	1,392,258
Coal required if old engine had performed this work.....	1,791,238 lbs.
Coal actually consumed	761,826 lbs.
Coal saved.....	1,029,412 lbs.

In January, 1896, two coal tests were made of the triple expansion vertical crank and fly wheel engine at Brookline, Mass., to determine the relative duty at different speeds.

It has been stated that a crank and fly wheel engine when run at a speed much slower than the contract requirements, shows a greater loss in economy than a direct acting duplex engine working under the same conditions. The test at Brookline, however, shows that with the conditions which existed at that station the performance of the engine in relation to economy in steam consumption was practically the same at the different rates of speed under which the engine was tested. The lower duty on a coal basis at the slower speed is explained in two ways: First, the coal burned per square foot of grate surface per hour in the second test was too small to give best results, and the water evaporated per pound of coal was $6\frac{2}{10}$ per cent. less. Second, the radiation of the boiler steam pipes and engine is nearly the same per hour at all rates of speed of the engine, and consequently the loss due to radiation forms a greater per cent of the work done as the work performed by the engine decreases.

Making corrections due to power, evaporation and increased loss of radiation, the duty of the engine was slightly better at the slower speed, being 140.3 million foot-pounds, and the actual duty of the faster speed being 140.1 million foot-pounds.

The test was a long one, lasting over twenty-eight hours, each, in order to make the error as small as possible due to more or less coal on the grates at the close of the test, water in the boiler, etc. Before the test all the gauges, and the scales for weighing the coal were tested and made correct. The data of the tests were all carefully taken at intervals of fifteen minutes. The results are appended, and I think will be of interest:

TEST NO. 1.

1. Duration of test—28 hours 26 minutes.
2. Total revolutions—50,276.
3. Revolutions per minute—29.5.
4. Total gallons pumped—5,982,844.
5. Total head against plungers—202.18 feet.
6. Total wet coal—7,200 pounds.
7. Coal per hour—253.3 pounds.
8. Moisture in coal—2.77 per cent.
9. Refuse in coal—6.63 per cent.

10. Total non-combustible—9.4 per cent.
11. Grate surface—25.0 square feet.
12. Coal per hour square foot of grate—10.13 pounds.
13. Duty per 100 pounds of wet coal—140,113,320 foot-pounds.

JACKET WATER, ETC.

14. Drain from high and intermediate cylinder jackets; first and second receiver jackets, and drain from separator on main steam pipe—8,708 pounds (returned by gravity to boiler).
15. Temperature of jacket returns near boiler—320° F.
16. Drain from low pressure cylinder jacket—2,032 pounds (drained into feed tank).
17. Drain from high pressure steam chest and bottom of high pressure cylinder—73 pounds 1 ounce per hour (drained into sewer through coil in feed tank).
18. Temperature of water pumped, main engine—50° F.
19. Temperature of water in feed tank—125° F.
20. Temperature of air pump discharge—108° F.
21. Temperature of gases in boiler uptake—418° to 511° F.
22. Steam pressure by gauge at engine—130 pounds.
23. First receiver—21.5 pounds.
24. Second receiver—minus 3.4 pounds.
25. Vacuum—13.2 pounds.
26. Coal used—George's Creek, Jackson Mine.

MANNER OF FIRING.

Coal spread on front of grate every 7½ minutes. Fire leveled off, furnace doors closed, and automatic damper allowed to nearly close one minute before each firing. Hand damper nearly closed before opening furnace doors. Coal sprinkled with warm water before using. Condition of fire, height of water in boiler, and steam pressure the same at beginning and end of test.

TEST NO. 2.

1. Duration of test—28 hours 45 minutes.
2. Total revolutions—37,089.
3. Revolutions per minute—21.5.
4. Total gallons pumped—4,413,591.
5. Total head against plungers—183.05 feet.
6. Total wet coal consumed—5,100 pounds.
7. Coal per hour—177.4 pounds.
8. Moisture in coal—2.77 per cent.
9. Refuse in coal—6.85 per cent.
10. Total non-combustible—9.62 per cent.
11. Grate surface—25 square feet.
12. Coal per hour per square foot of grate—7.1 pounds.
13. Gallons pumped per pound of wet coal—865.4 gallons.
14. Duty per 100 pounds wet coal—132,116,693 foot-pounds.

JACKET WATER, ETC.

15. Drain from high and intermediate cylinder jackets; first and second receiver jackets, and drain from separator on main steam pipe—6,840 pounds (returned by gravity to boiler).
16. Drain from separator only—672 pounds.
17. Drain from low pressure cylinder jacket—1,713 pounds.
18. Water returned from jackets and separator—6,840 pounds.
19. Total feed water—50,965 pounds.
20. Steam condensed in jackets—7,831 pounds. (15.464 per cent).
21. Temperature of water pumped—50°.
22. Temperature of water in feed tank—114°.
23. Temperature of discharge from air pump—98°.
24. Steam pressure (by gauge) at engine—130 pounds.
25. First receiver—16.5 pounds,
26. Second receiver—minus 3.5 pounds.
27. Vacuum—13.6 pounds.
28. Coal used—George's Creek, Jackson Mine.

THE POSSIBILITIES OF ECONOMY IN PUMPING ENGINES AS BASED ON THE LATEST ACCOMPLISHMENTS.

BY GEO. H. BARRUS, M. E., BOSTON, MASS.

[*Read Dec. 14, 1898.*]

The highest interest in the possibilities of the performance of pumping engines has been awakened by the results obtained from a 20,000,000 gallon engine built by the Snow Steam Pump Works of Buffalo, N. Y., for the Indianapolis, Water Co. According to a duty trial made by Prof. Goss of Purdue University, this engine gave a duty of 150.1 million foot-pounds, based on a consumption of 1,000,000 heat units, and 167.8 million foot-pounds based on a consumption of 1,000 pounds of dry steam. These results, are, I believe, higher than any heretofore published, and are naturally viewed by engineers and users of pumping engines with deep interest.

In the following table a comparison is made between the essential features in the performance of this engine, with those of a number of other prominent engines which have been carefully tested within the past five years, together with some data and calculations made by the writer not found in the published reports of the tests; and I propose in this paper to present a brief examination of these trials, with a view to showing what it is reasonable to expect of the duty of a pumping engine under the best obtainable conditions.

The data given in this table for the Milwaukee engine are taken from Vol. XV., Transactions of the American Society of Mechanical Engineers; that for the Chestnut Hill engine from Vol. IX., Technology Quarterly; that for the Detroit and Buffalo engines from the original official reports, and that for the Indianapolis engine from a typewritten copy of Prof. Goss' report, furnished by Mr. Davis, president of the Indianapolis Water Co.

1. Name of designer or builder . .	E. P. Allis Co.	E. D. Leavitt, Jr.	E. P. Allis Co.	Lake Erie, Eng. Works.	Snow Steam Pump Works.
2. Locality	Milwaukee, Wis.	Chestnut Hill, Mass.	Detroit, Mich.	Buffalo, N. Y.	Indianapolis, Ind.
3. Type	Trip. Ex.	Trip. Ex.	Trip. Ex.	Trip. Ex.	Trip. Ex.
4. Extent of jacketing	Barrels and receivers.	Barrels, heads and receivers.	Barrels and receivers.	Barrels and receivers.	Barrels, heads and receivers.
5. Name of expert conducting test .	Prof. R. C. Carpenter.	Prof. E. F. Miller.	Geo. H. Barrus.	Geo. H. Barrus and Newcomb Carlton.	Prof. W. F. M. Goss.
6. Capacity—million gals. in 24 hours	18.	20.	24.	30.	20.
7. Size of steam cylinders, in. . . .	28, 48, 74 x 60	13.7, 24.37, 39 x 72	28, 48, 74 x 60	37, 63, 94 x 60	29, 52, 80 x 60
8. Size of water plungers, in. . . .	32 x 60	Double acting. 17.5 x 48	36 x 60	42 x 60	33 x 60
9. Total head, lbs.	70.4	59.4	53 4	86.1	88.7
10. Piston speed feet per minute . .	203.1	607.	209.9	207.7	214.6
11. Ratio of volume of L. P. cylinder to volume of H. P. cylinder .	7.1	8.3	7.1	6.5	7.7
12. Pressure near throttle(above atmosphere) lbs	121.4	175.7	125.2	167.1	155.6

13. Cut-off pressure (above atmosphere), lbs.	118.6	151.5	119.4	152.2	153.
14. Release pressure, L. P. cylinder (above zero), lbs.	5.3	6.9	5.8	7.4	6.4
15. Back pressure L. P. cylinder (above zero), lbs.	1.6	1.5	2.8	2.2	2.5
16. Cut-off, H. P. cylinder	.337	.384	.338	.323	.315
17. Clearance, H. P. cylinder	.014	.015	.014	.014	.018
18. Ratio of expansion	20.4	21.	20.3	19.6	23.8
19. Ratio of expansion referred to pressure near throttle	20.8	23.9	21.2	21.3	24.3
20. Absolute pressure near throttle ÷ ratio of expansion, lbs.	6.6	9.1	6.9	9.3	6.8
21. Indicated horse power, I. H. P.	573.9	575.7	573.7	1185.5	775.5
22. Friction, %	9.2	10.5	10.2	5.1	4.6
23. Dry steam per I. H. P. per hour, including jacket and reheater steam, lbs.	11.68	11.22	12.52	12.39	11.26
24. Per cent of steam condensed in jackets and reheaters, %	9.2	17.1	12.7	13.7	10.5 (est.)
25. Dry steam per I. H. P. per hour, exclusive of steam used in jackets and reheaters, lbs.	10.61	9.3	10.93	10.7	10.08 (est.)

26. Steam accounted for by indicator, H. P. cylinder cut-off, lbs. . .	9.05*]	8.5*	9.5	9.1	8.7*
27. Steam accounted for by indicator, L. P. cylinder cut-off, lbs. . .	8.7*	9.6*	9.5	9.7	8.7*
28. Steam accounted for by indicator, L. P. cylinder release, lbs. . .	9.04*	9.1*	9.9	9.7	9.8*
29. Cylinder condensation and leakage, including jacket and reheater condensation, at cut-off, H. P. cylinder	.225	.242	.241	.266	.227
30. Mean effective pressure referred to L. P. cylinder, lbs.	21.77	26.36	21.03	27.19	23.65
31. Theoretical mean effective pressure referred to pressure near throttle valve and ratio of expansion at same point	26.42	33.23	26.76	34.63	29.66
32. Line 30 ÷ line 31, or "diagram factor"	.824	.794	.792	.786	.797
33. Duty based on 1,000,000 heat units, expressed in million foot-pounds	137.	141.9	129.7	135.4	150.1
34. Duty based on 1,000 lbs. of dry steam, expressed in million foot-pounds	154.0	154.9	142.4	152.	167.8

*Calculated by G. H. B.

I have carefully examined the reports and records of these tests, and I believe them all to be substantially reliable. In the report of the Indianapolis test, Prof. Goss states that the indicator diagrams, owing to long driving cords, are not altogether satisfactory. The diagrams are, on this account, of different lengths, it is possible that the indicated horse-power is greater than the amount reported and the friction greater by a corresponding amount; but it should be observed that whatever increase there may be in this direction, makes the steam consumption per indicated horse-power per hour correspondingly less. As this, as it stands, is an exceptionally low quality, the error, if any, must be a small one. I understand that this test has just been repeated, and Prof. Goss reports that the results obtained on the first test have been completely verified.

With the exception of the two tests conducted by the writer, there is no published statement as to the condition of the valves and pistons of the steam cylinders as regards tightness, and the precise conditions in this respect must be inferred from the character of the results. As to the Detroit and Buffalo engines, the valves and pistons were tested for leakage with the engines at rest, subjected to the working pressures; and in neither case was the engine found to be in the best condition. Both of these engines leaked to some extent.

The economy with which an engine uses steam, assuming a proper ratio of expansion, is dependent upon three things: First, the steam pressure; second, the efficiency with which the theoretical expansive force of the steam is realized, as shown by the proportion which the area of the actual indicator diagram bears to the theoretical diagram; and third, the quantity of cylinder condensation and leakage. An increase in the steam pressure, an increase in the proportion of the theoretical expansive force realized in the actual diagram, and a decrease in the quantity of cylinder condensation and leakage, all tend toward greater economy of the engine; that is, to a reduction in the quantity of steam used in producing a given amount of power. As applied to pumping engines in which the economy is measured by the amount of water pumped by a given unit of energy (such as the energy produced by the combustion of 100 pounds of coal, or the consumption of 1,000 pounds of steam, or by the use of 1,000,000 heat units), the economy is

dependent, also, upon the efficiency of the mechanism of the engine and pump, that is, upon the friction; the smaller the friction, the greater the amount of useful work done by the given unit of energy. That a proper comparison may be made of the results of the tests of these five engines, the data and calculations showing these four controlling factors are here presented. Line 12 gives the pressure; line 32, the diagram factor, or the proportion which the actual mean effective pressure measured from the diagrams bears to the theoretical mean effective pressure; line 29, the cylinder condensation and leakage at cut-off H. P. cylinder (including condensation in jackets and reheaters); and line 22, the friction, this applying to the entire mechanism and the friction of the water between the pump well and the discharge main.

In the Milwaukee engine the pressure is 121.4 pounds, which is low compared with most of the other engines; the diagram factor is .824, which is the highest of all; the cylinder condensation and leakage at cut-off H. P. cylinder is .225, which is the lowest of the whole list; the friction is 9.2 per cent., which is comparatively high; and the duty based on heat units is 137,000,000.

In the Chestnut Hill engine the pressure is the highest of all the engines given in the table, being 175.7 pounds. The diagram factor, however, is with one exception the lowest given in the table, being .794. The cylinder condensation and leakage at cut-off H. P. cylinder is .242, and the friction, 10.5 per cent., both of which are comparatively high. The duty in this case is 141.9 millions on the heat unit basis.

In the Detroit engine the pressure is 125.2 pounds; the diagram factor is .792, which is the lowest of all; the cylinder condensation and leakage at cut-off H. P. cylinder is .241, and the friction is 10.2 per cent.; all of which are medium results compared with their fellows.

In the Buffalo engine the pressure is 167.1 pounds, which is nearly the highest of the whole list; the diagram factor is .786; the cylinder condensation and leakage at cut-off H. P. cylinder is .266, which is larger than any other figure of the whole five engines; and the friction is 5.1 per cent., one of the lowest. The resulting duty, based on heat units, is 135.4 millions.

In the Snow engine the pressure is 155.6 pounds, which is a medium figure compared with the others; the diagram factor is

.797; the cylinder condensation and leakage at cut-off H. P. cylinder is .227, which with one exception is the lowest of all; and the friction is 4.6 per cent., which is smaller than any other on the list. With substantially one exception, all these quantities are the most favorable to high economy of any in the table.

In the light of what this analysis reveals, a duty of 150,000,000 foot-pounds of work for 1,000,000 heat units is a result which ought to be readily duplicated, and very likely surpassed, by any of the various types of engines here referred to, provided they are given an equally favorable opportunity. Let us go over the list and see what the possibilities are in the case of the engines which gave the lower duties, provided the conditions referred to were more favorable. The Milwaukee engine worked under a steam pressure of 34.2 pounds below that of the Indianapolis engine. I estimate that if the pressure had been raised in the Milwaukee engine to the same point, with a corresponding increase in the ratio of expansion, the duty would have been increased 7.1 per cent. Furthermore, if the friction of the Milwaukee engine had been no greater than that of the Indianapolis engine, the duty would have been still further increased 5.1 per cent. These two changes would have brought the duty per 1,000,000 heat units, up to 154.2 million foot-pounds. Again, if the Chestnut Hill engine had operated with the same cylinder condensation and leakage as the Indianapolis engine, the duty would have been increased 2 per cent.; and, further, if the friction had been no greater in one case than in the other, the duty would have been increased 6.6 per cent. more, bringing the final result up to 154.2 millions. In the Detroit engine an increase in the boiler pressure from 125.2 pounds up to 155.6 pounds, with a corresponding increase in the ratio of expansion, would have increased the duty, according to my estimates, to the extent of 8.2 per cent. If the cylinder condensation and leakage had been no greater in one case than in the other, the duty would have been further increased 1.9 per cent. If, also, the friction had been only 4.6 per cent., there would have been a still further increase in the duty of 6.5 per cent., making the final result 152.3 millions.

The possibilities of economy in pumping engines, as based on the latest accomplishments, in a word, seem to be that with a boiler pressure of 175 pounds per square inch, a diagram factor, cylinder condensation and leakage as favorable as obtained in the Milwaukee

engine, and the friction as low as 5 per cent., which appears to be capable of realization, a duty can be obtained of at least 155,000,000, and possibly 160,000,000 foot pounds, for the expenditure of 1,000,000 heat units.

DISCUSSION.

THE PRESIDENT. The paper is now open for discussion, and we would like to hear from anyone present, particularly from the mechanical engineers, quite a number of whom are here. I will call upon Mr. F. W. Dean.

MR. DEAN. Mr. President, some years ago at the Worcester meeting I read a paper before this Association on Pumping Engines, and I thought to-day I would come to listen and not to talk. Of course these records of performances are very interesting, and I think we all ought to be grateful to Mr. Barrus for collecting the records in this shape, which is convenient to file away.

The question of the possible duty of a pumping engine is something which is occurring to people all the time, and I think we have the right to expect that even this high duty of the Snow engine will be exceeded in a few years. It is within the range of possibility that steam consumption may come down another quarter of a pound, and I do not know but there may be people who will be willing to guarantee that it will. I have been told that the Edward P. Allis Company would guarantee 11 pounds of steam; I don't know whether that is so or not, but I hope we will find out from Mr. Reynolds. Of course when you get down to small results quarter pounds count, and they are equally difficult to drop off from the result. I received quite recently a catalogue of engines from the Wheelock Engine Co., of Worcester, in which they state on the fly-leaf that they are prepared to guarantee a consumption of one pound of coal per indicated horse power per hour. Now, I do not deny that the consumption can be brought down to that point, for it is barely possible, in fact I know it is possible, to get an actual evaporation from a pound of coal of something over 11 pounds. I did that with two tests on a New Bedford water works boiler myself, just a shade over 11, and with the feed-water of 130 degrees, I believe. Now, if the feed-water by some means or other could have been brought up to 200 degrees, there would be 70 degrees more, which would mean about 7 per

cent. of coal saved, and the resulting boiler performance connected with one of these highly economical engines, I have no doubt would have brought the coal consumption down to one pound. So, while it is a very hazardous thing to make any such guarantee as that, it is not an impossibility. Anybody who has had experience in obtaining guarantees in boiler performances, knows how hazardous it is to meet the guarantee with them. While one can quite safely guarantee the perfoamance of an engine, if he is careful to see everything in it is tight, when you come to a boiler, no matter how good the boiler is, you always have present the means of the poorest possible performance in the world, even if the boiler is the best one in the world. I have myself quite recently had a terrific struggle to get one of the best boilers in the world to give anything better than the poorest result. It was finally done, however, by finding a man who happened to have the right kind of brains to enable him to put in his coal properly and watch his fire properly.

PRESENT PUMPING ENGINE PRACTICE OF THE EDWARD P. ALLIS CO. COMPARED WITH THAT OF TWENTY-FIVE YEARS AGO.

BY IRVING H. REYNOLDS.

[*Read Dec. 14, 1898.*]

The great advance in the economy of pumping engines during the past twenty-five years, and the decrease in cost of high duty engines during this period, may perhaps render of interest to you an engineer's review of the causes which have brought about this dual result.

Twenty-five years ago, 75,000,000 duty was a high guarantee, and 100,000,000 the maximum that had been obtained, while today 150,000,000 duty is guaranteed, and 160,000,000 duty appears within reach if not already attained. The cost of high duty engines twenty-five years ago was for ordinary heads approximately $10,000 per million gallons capacity while today machines 50 per cent. more economical can be purchased for one-third of this price.

The work of the Edward P. Allis Company illustrates very well the increase in economy and decrease in cost of pumping engines during the past quarter of a century, but before going into details of this I wish to mention some of the pioneer work of others.

The compound engine, which has been the greatest factor in increasing the economy, appeared as a pumping engine early in the seventies, and I believe Mr. Leavitt's Lynn engine was the first machine to give 100,000,000 duty on coal, the Lowell engine, designed by Mr. Morris, built just previously giving a duty of 94,000,000.

The introduction of the compound engine was opposed by many, as was the introduction of the triple twenty years later, and so prominent an engineer as Geo. H. Corliss, whose name is identified with the progress of steam engineering, was bitterly opposed to the compound engine, and constructed the remarkable Hope street engine at Providence about 1873, making it a simple condensing

machine. The result was that at one-half speed (contract conditions) it gave a duty of only about 25,000,000 foot-pounds, while a compound Worthington machine, tested at the same time, gave a duty under similar conditions of over 50,000,000.

After the failure (as to economy) of the Hope street engine, Mr. Corliss built another simple condensing engine (which still stands in the Corliss works), which was to finally demonstrate that there was nothing to be gained by compounding. This was a comparatively high speed Corliss engine geared to a ten plunger pump, and with this he succeeded in getting a duty of about 60,000,000. After this he built an experimental compound pumping engine which was operated in his shops, and many years later was sold to Easton, Pa., and in 1877 he sold his first compound pumping engine, the now justly celebrated Pawtucket machine, which held the compound duty record for many years.

It is interesting to note that the tests of Mr. Corliss' experimental engines were conducted by the now well known yacht builder, Mr. N. G. Herreshoff, and the valve gear and some other details of the Pawtucket engine were designed by him.

The above is sufficient to show that there existed a difference of opinion as to the merits of compound engines.

The Allis Company built their first pumping engine in 1873, it being a double machine of 16,000,000 gallons capacity. The engine, which was designed by Mr. R. W. Hamilton, was one of the over-head beam type, and similar in general style to engines in the Brooklyn Water Works, and also in the London (Eng.) works, excepting that it was compound, while the others were not. This engine cost, with its boilers, about $165,000, or at the rate of over $10,000 for each million gallons capacity, and was guaranteed to develop a duty of 60,000,000 foot-pounds for each hundred pounds of anthracite coal, and on a 48-hour test gave a duty of about 76,000,000.

In 1881 the Allis Company built their second pumping engine, designed by Edwin Reynolds, which was also a compound vertical beam engine having a capacity of 12,000,000 gallons in 24 hours, and costing about $65,000, or at the rate of $5,400 per million gallons capacity. This engine with 80 pounds steam pressure developed a duty of nearly 105,000,000 per 100 pounds of coal.

In 1891 an 18,000,000 gallon triple expansion pumping engine

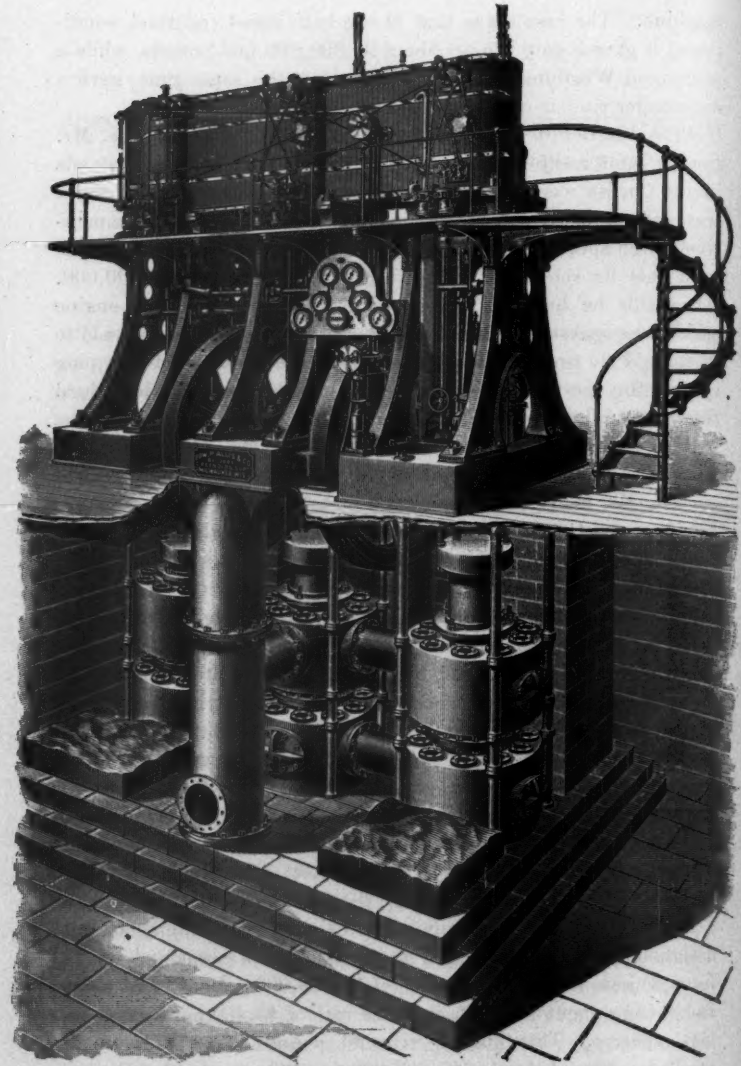


Fig. 1. The Original Triple Expansion Pumping Engine.
Built in 1886 by The Edw. P. Allis Co.

(see Fig. 4) was placed in the Milwaukee Water Works, which cost, together with its boilers, $76,000, or at the rate of $4,222 per million gallons capacity, the guaranteed duty being 125,000,000 foot-pounds per 100 pounds of anthracite coal. It will be seen that this engine, though of greater capacity, cost less than one-half the price of the original 16,000,000 gallon engine placed in the same station eighteen years before. The duty guaranteed was double, and in addition the space occupied was only about one-half that required by the earlier machine, and the cost of foundations was considerably less than half. In brief, eighteen years' progress of one builder represented equal capacity with double the duty at one-half the cost. I shall have occasion to speak of the performance of this last engine a little further on.

Between 1881 and 1886 the Allis Company built sixteen compound pumping engines of various types, some of them giving duties as high as 107,000,000 foot-pounds per 1,000 pounds of steam.

Nearly all of the earlier crank and fly-wheel pumping engines were of the beam type, in fact this type of engine was so universal that it almost seemed as though it was considered one of the fundamental requirements, and until 1886 all of the Allis pumping engines with the exception of two engines (Figs. 2 and 3) were of this type, but since the introduction of the triple in 1886 it has been our almost universal practice to build direct connected engines, avoiding entirely all forms of beams, bell cranks, or levers. We have settled down to practically three designs—the vertical triple expansion, the vertical compound, and the horizontal compound types.

A brief mention of the origin and development of the triple expansion engine will not be out of place here.

In 1886 the Allis Company built from the writer's designs, the first triple expansion pumping engine. (Fig. 1). This engine, which was constructed for the high service station in the Milwaukee Water Works, was the first triple expansion pumping engine built in America, and as far as I can learn the first to be designed by anyone, although an engine was constructed at about the same time for the East London (Eng.) Water Works.

It is a rather remarkable coincidence that two designers, totally unacquainted with each other or with the other's work, should at

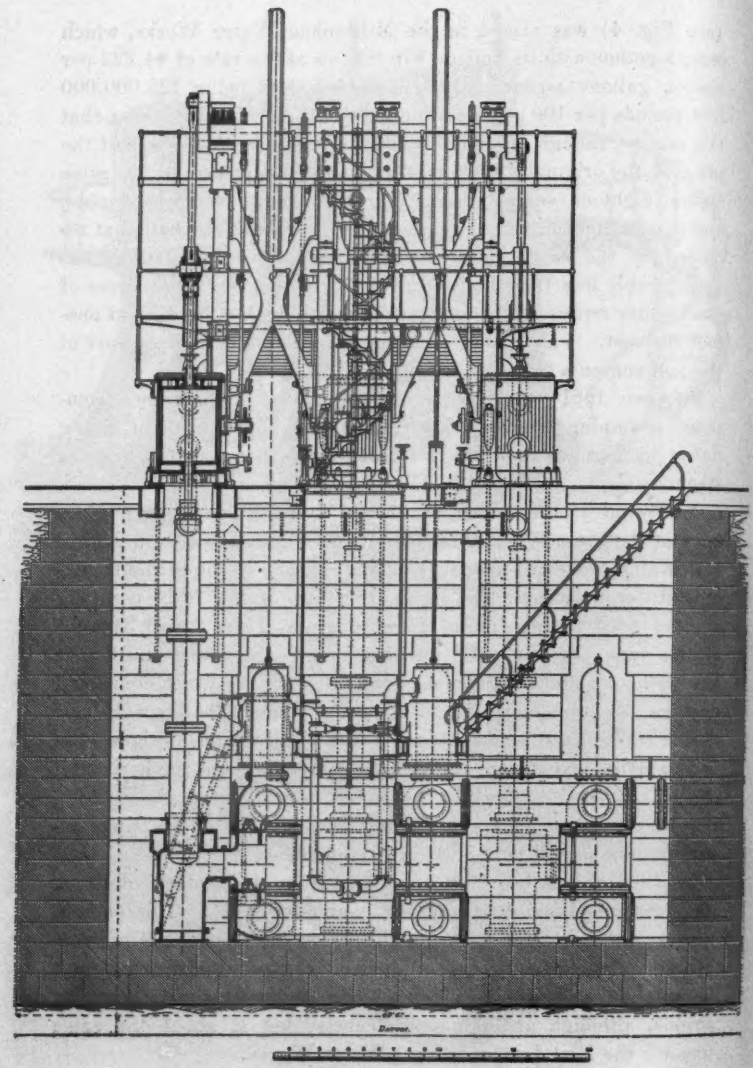


Fig. 2. Side Elevation.
THREE CRANK COMPOUND ENGINES.
Built 1883.

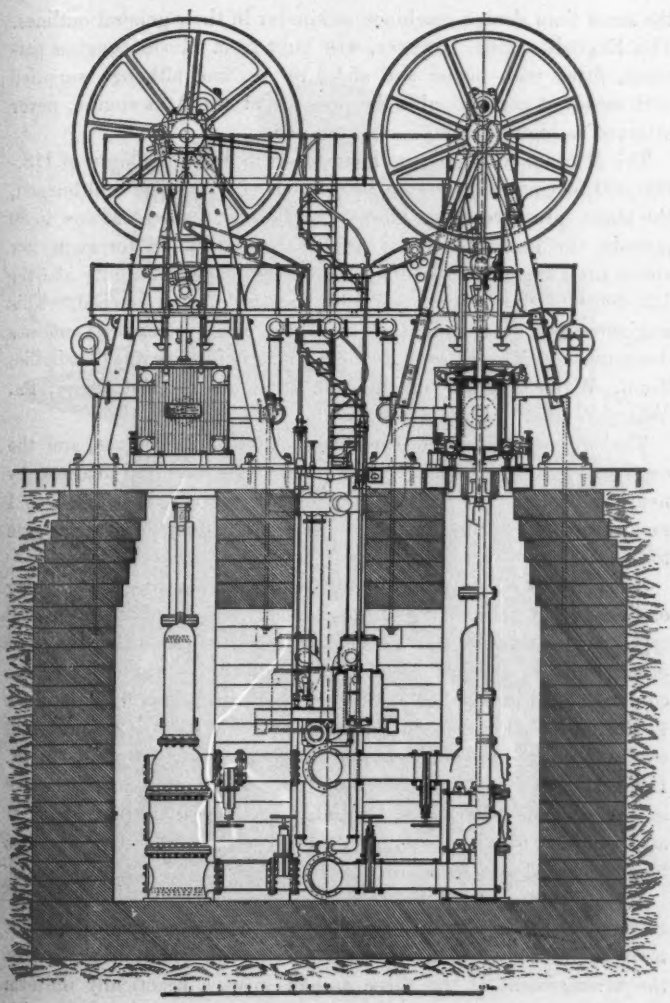


Fig. 3. End Elevation (2 Engs.)
THREE CRANK COMPOUND ENGINES.
Built 1883.

the same time design machines so similar in their general outlines. The English engine, however, was built from Marine engine patterns, fitted with piston and slide valves, and although supplied with steam at nearly double the pressure of the Allis engine, never attained as high economy.

The Milwaukee triple was guaranteed to develop a duty of 118,000,000 foot-pounds for each 100 pounds of anthracite coal burned, the steam pressure being limited by the city's specifications to 80 pounds, the guarantee being unprecedentedly high for such low steam pressure. But the engine actually developed a duty of over 125,000,000 foot-pounds per 100 pounds of coal burned. This engine was a development of the three crank compound engines designed by Mr. Edwin Reynolds originally for the Nashville, Tenn. Water Works, but built in 1883 for the Allegheny, Pa. Water Works. (Figs. 2 and 3).

The triple expansion engine was an immediate success, and the original engine has been copied in all of its essential features by nearly all builders of high duty pumping engines, and I believe I may safely say is the standard type of high duty pumping engine today.

The Allis Company has built 45 triple expansion pumping engines, 33 of them being of the vertical and 12 of the horizontal type, and engines of this general type, by the Allis Company and other builders, are in use in all of the important cities of the country. Milwaukee has 3, Chicago 6, Philadelphia 6, New York 4, Boston 5 under construction (besides the Leavitt Chestnut Hill engine, which has been running several years); Buffalo 2, St. Louis 4, (and 5 more in contemplation), while other machines are in New Orleans, Cleveland, Detroit, etc. The above list is not at all complete, but I believe is sufficient to show that the type has been accepted as a standard.

I am an advocate of the triple expansion engine, not solely because of the economy due to the triple expansion steam end, but more particularly because it is a triplex pump, which on account of the arrangement of the three cranks, gives a practically uniform flow of water with pumps of very simple construction. For this reason I consider the triple expansion engines which have say two plungers, and either three or four steam cylinders, but little better than compound engines, for while they undoubtedly effect some

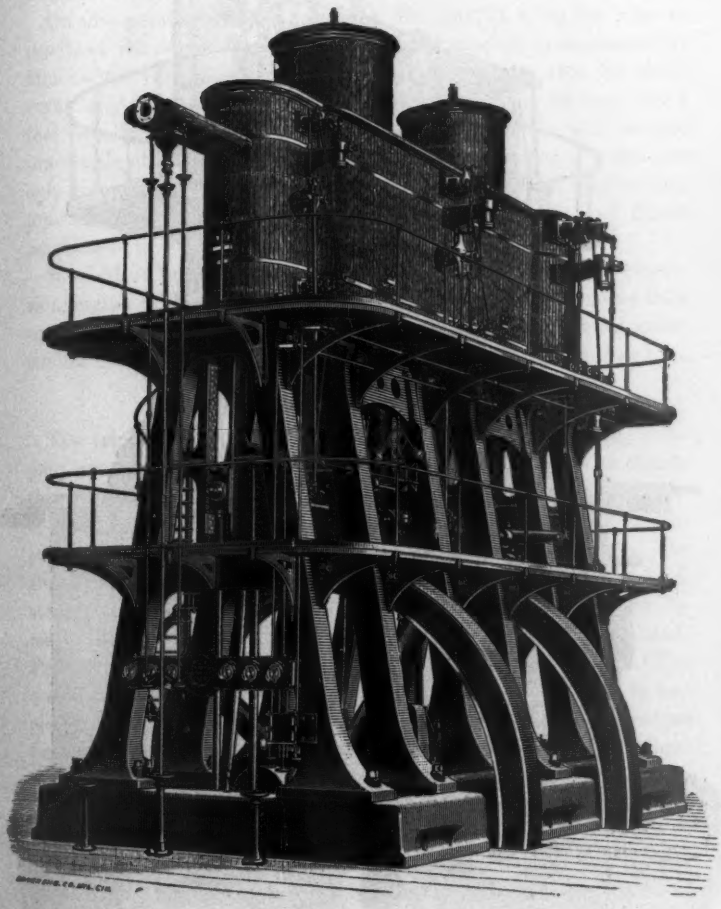


Fig. 4.

TRIPLE EXPANSION PUMPING ENGINE.

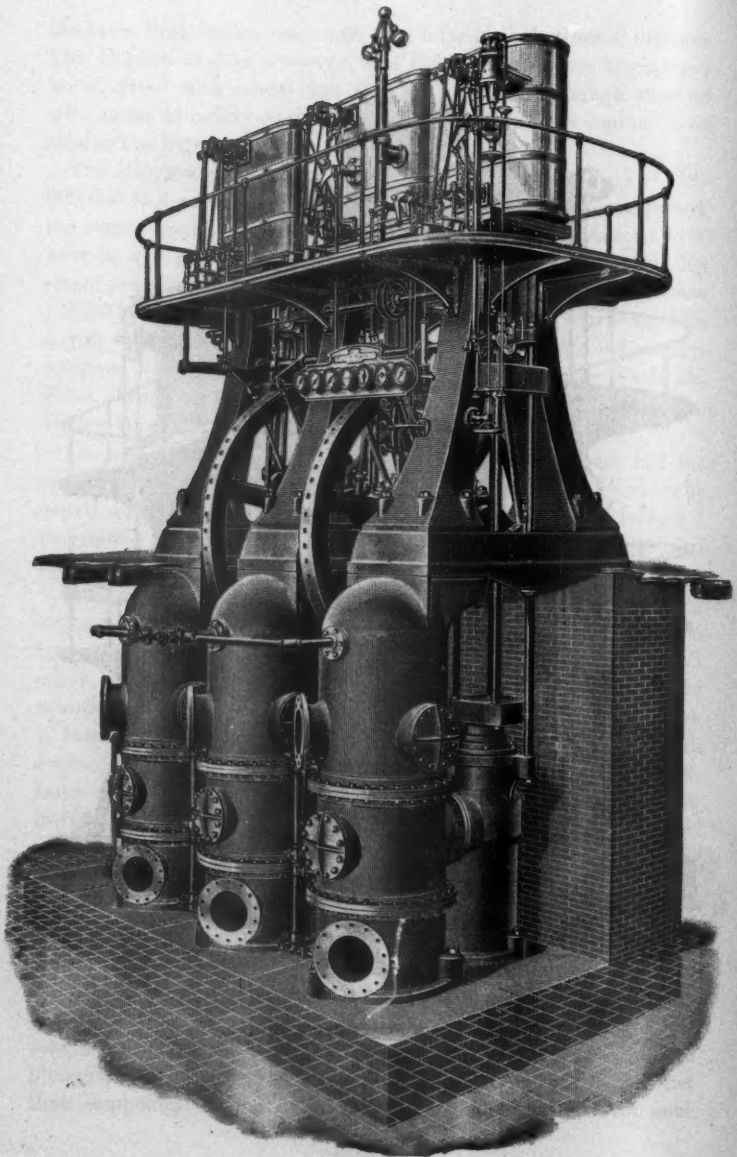


Fig. 5.
THE REYNOLDS TRIPLE EXPANSION PUMPING ENGINE.

gain in economy, the delivery of water is no better than that of the compound engine.

On this point I cannot agree with Mr. Leavitt as to the relative merits of the single crank and three crank engine, as expressed by him before your Association in 1894, for I believe that the three crank engine, solely because of its merit, has come to stay, and I feel equally sure that the single crank beam engine will become practically obsolete in the near future. This opinion is based not only on pumping engine practice, but on general steam engineering practice the world over, which tends towards simple and direct machines.

In addition to its superior mechanical features, the triple expansion engine will give from 10 to 15 per cent. higher economy than compound engines of the same general construction working under similar conditions, even with steam pressures of 100 pounds or less.

TRIPLE EXPANSION ENGINE.

The triple expansion pumping engines built by the Allis Company are so well known, and the illustrations show them so clearly that the following brief description is hardly necessary:

These engines are of the vertical 3 cylinder type, similar in general outline to the ordinary triple expansion marine engine. The steam cylinders are mounted on top of cast iron "A" frames, each steam piston being attached directly to its own pump plunger by means of piston and distance rods connecting to a cross-head. From this cross-head extends the connecting rod to crank pin. The crank shaft has three cranks set at 120°, the shaft being in two pieces, with the center crank fitted with drag boxes to prevent breaking or straining the shaft in case the engine gets out of alignment. There are two fly-wheels on the shaft to secure a steady turning effect, although with a three crank engine it is possible to run with very light ones, or even without wheels.

Corliss valve gear is fitted to all of the cylinders of the smaller engines, the larger engines having Corliss valves on the high and intermediate cylinders, and poppet valves on the low pressure cylinder.

The steam cylinder barrels are jacketed, and the steam and exhaust valves are placed in the cylinder heads.

The engine is fitted with a fly-ball governor and also hand cut-off

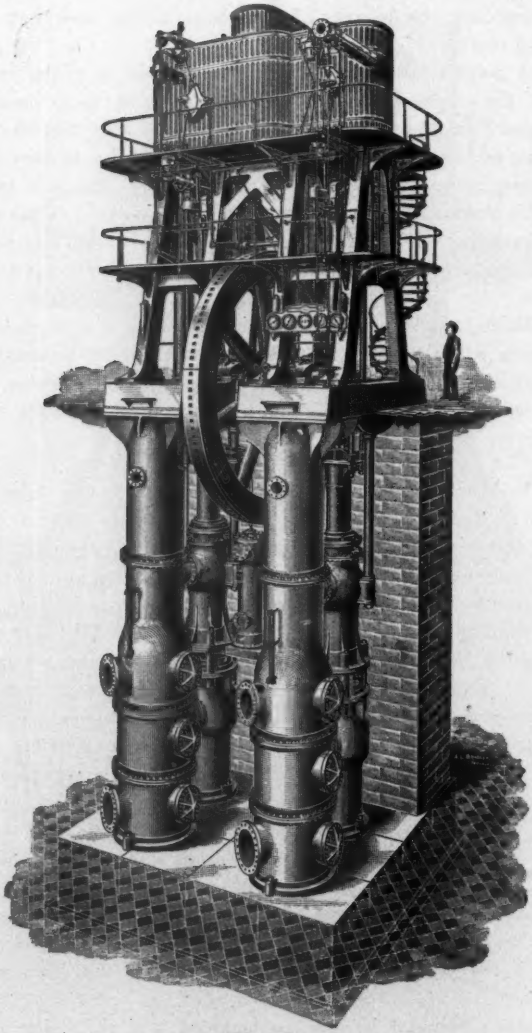


Fig. 6.

FOUR VERTICAL COMPOUND PUMPING ENGINES.
Built for Pittsburg, Pa. Cylinders 50" and 92" diam. x 64" stroke.

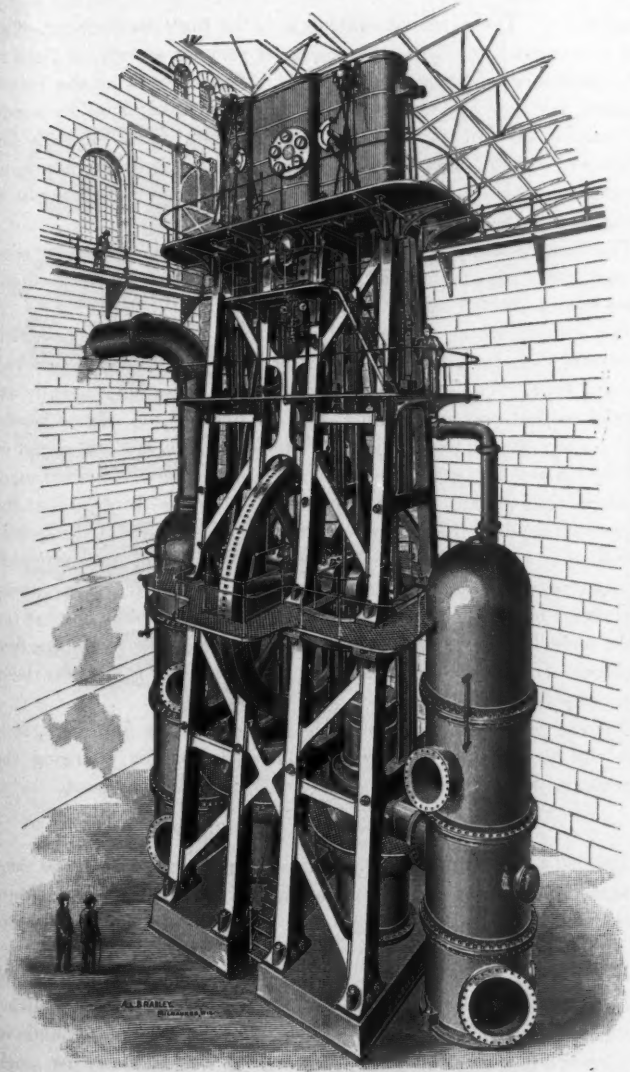


Fig. 7.

FOUR VERTICAL COMPOUND PUMPING ENGINES.

For St. Louis, Mo. Cylinders 27" x 52" diam. x 108" stroke.

attachment. The governor controls only the high pressure cut-offs, and is arranged to control the engine at varying speeds, or limit it at a predetermined maximum speed. The cut-offs on the intermediate and low pressure cylinders are always adjusted by hand, and require no adjustment for changes in load, unless possibly for extremely wide changes where, for instance, fire pressure is put on the engine which is double the normal pressure, in which case a slight lengthening of the cut-offs would be desirable.

The throttle valve, cut-off adjustments, injection valves for condenser, etc., are all grouped together so that even the largest engines can be started by one man without difficulty.

The pumps, which are located beneath and directly in line with the steam cylinders, are usually of the single acting outside packed plunger type. The pumps are cylindrical in form, and usually arranged so that one end of the main bed plate of the engine is carried on top of the pump chambers (see Fig. 5), the other end of the bed plate being supported by a masonry pier. This arrangement reduces the cost of the foundation about one-third, and at the same time renders the pumps much more accessible. It has sometimes been criticised on account of the supposed difference in expansion between the masonry pier and the cast iron in the pump chambers, but the majority of our engines built during the past ten years have been arranged this way, and I have yet to see the first evidence of any difference in expansion, nor has there ever been any complaint on this score.

The air pump is driven directly from one of the plungers, as is also the feed pump and small air compressor for charging the chambers, so that there are no auxiliary pumps of any kind.

VERTICAL COMPOUND ENGINE.

The Allis vertical compound engine (Fig. 6) is built on the same general lines as the vertical triple, and is practically two-thirds of that machine, one cylinder, frame, bed, pump, etc., being omitted. Where it is necessary to use compound engines on direct service work, we ordinarily set the crank at 90°, and use differential plunger pumps in order to minimize the pulsation on the delivery main, although this can be accomplished by the use of air chambers. For reservoir work with compound engines we ordinarily use single acting plungers with the engine cranks set at 180° (Fig. 7). This

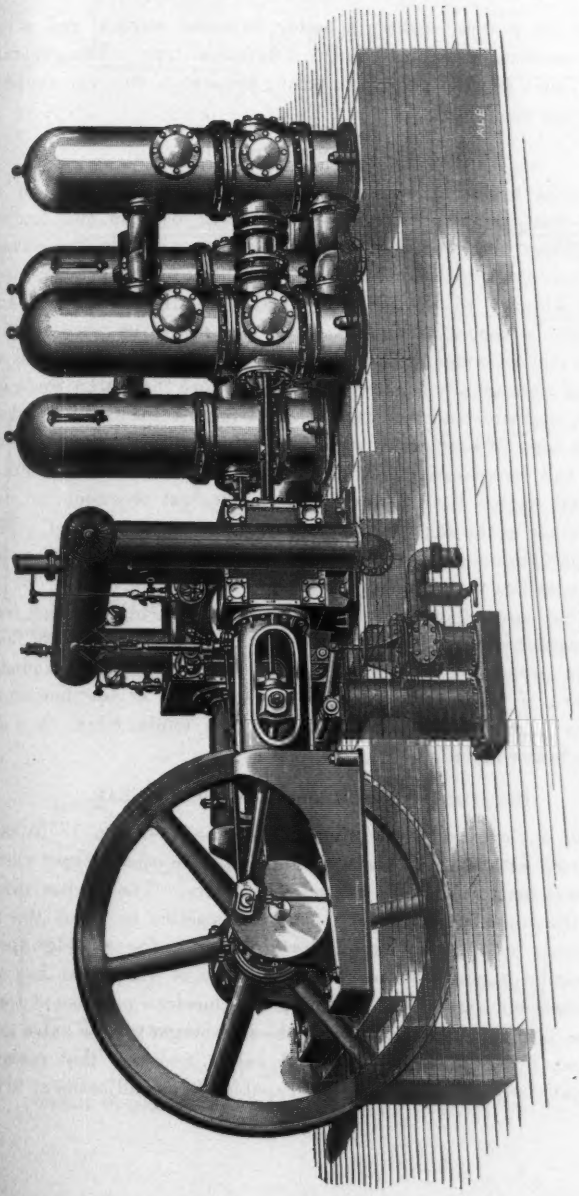


Fig. 8.

HORIZONTAL COMPOUND PUMPING ENGINE.

simplifies the pumps, making a better balanced engine, and is in many ways more desirable than the differential type. The general features, such as valve gear, condensing apparatus, etc., are similar to the triple machine.

HORIZONTAL COMPOUND ENGINE.

The Allis horizontal compound pumping engine consists of a pair of cross-compound Corliss engines (Fig. 8) with two double acting pumps arranged tandem to the steam cylinders, the steam piston rods being extended through the back cylinder heads and coupled directly to the plunger rods. The engine cranks are set at 90°, which, with double acting pumps, gives practically a uniform flow of water. The pumps are usually the outside packed type, but occasionally for low pressures, where the water contains no sand, inside piston pumps are used. I regard this as the simplest and best form of horizontal pumping engine, as it is direct, easy of access, and inexpensive, and nothing more can be desired where a horizontal engine is to be used. The principal objection to this machine lies in the relatively large amount of space required. The horizontal engine is not, however, recommended excepting for small capacities under moderate heads.

The horizontal triple is not recommended excepting under very unusual circumstances, for it occupies a very large amount of floor space in proportion to its capacity, and when the cost of foundation, etc., is taken into account, the total cost of the horizontal triple is nearly as much as for the vertical triple, which is a decidedly better machine.

DETAILS OF ENGINES—STEAM VALVE GEAR.

All of our engines, (excepting the first pair built in 1873) have been fitted with Corliss valve gear, and in some cases poppet valves have been used on the low pressure cylinders. The Corliss valve, while theoretically imperfect, seems in practice to meet the requirements for all steam engine work (excepting for very high speed and marine engines) better than any type of valve that has yet been designed, and is fast becoming in American practice at least, the standard valve gear. Some of the advantages of this valve gear are that it has a nearly balanced valve, and one that remains practically steam tight. It is also capable of easy adjustment to se-

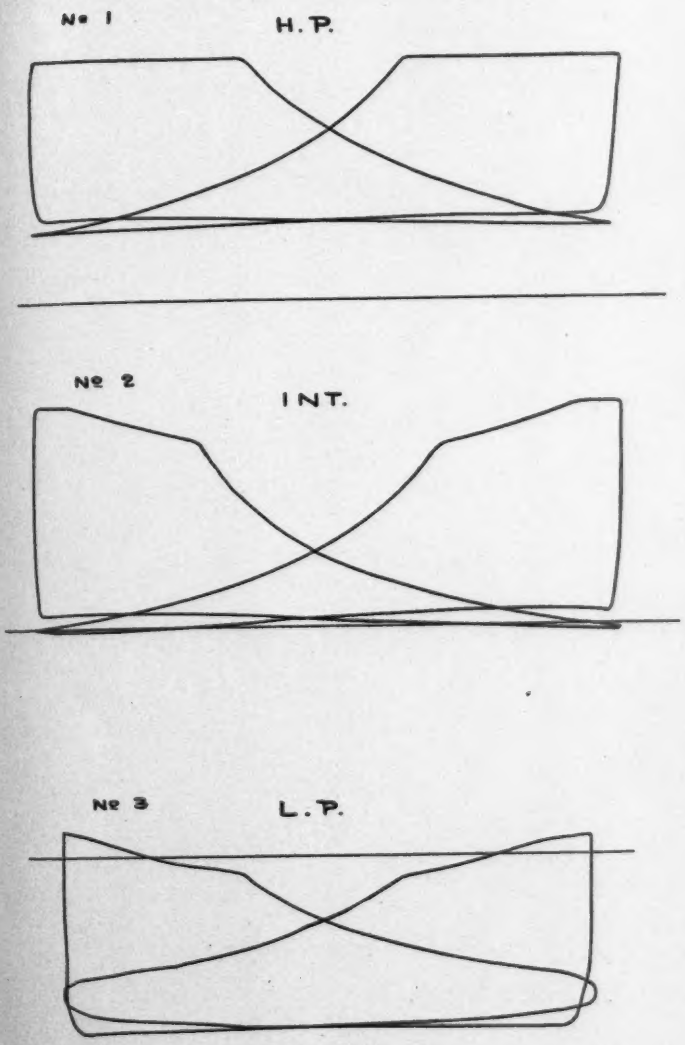


Fig. 9.

STEAM DIAGRAMS, FROM TRIPLE EXPANSION PUMPING ENGINE.

cure proper steam distribution, (Fig. 9) and on pumping engines can be adjusted while the engine is running. The valves are also easily removed without disturbing the setting of the valve gear.

The Corliss valve gear is essentially one in which the steam and exhaust valves are separated, i. e., each cylinder having two steam and two exhaust valves, and the mis-called Corliss valves, applied to some of the direct acting pumps, are merely rolling slide valves, which, being unbalanced, wear badly and require a great deal of lubrication.

It is now our standard practice on vertical engines, to place the valves in the cylinder heads, first with the object of reducing the port clearance, and secondly, to secure a simple form of steam jacketed cylinder. On the largest vertical engines we use poppet exhaust valves on the low pressure cylinder, and occasionally poppet steam valves on the low as well as poppet exhausts on both the intermediate and low. The use of poppet valves is for a two-fold purpose: First, to reduce the clearance to a minimum (⅜ of 1 per cent. in some instances), and secondly, to secure an absolutely tight valve, which would be very difficult with the light receiver pressure, and the large Corliss valves that would be necessary for 90-inch or 100-inch low pressure cylinders.

PUMP CONSTRUCTION.

Vertical pumps we usually build of the single acting outside packed type, as this is the simplest form, and as it has only one set of suction valves there is no difficulty with the pumps filling uniformly, while with double acting vertical pumps, particularly of long stroke, (see Fig. 7) the upper suction valves being usually several feet above the lower ones, the upper end of the pump may not fill as perfectly as the lower, particularly where there is a high suction lift.

Differential pumps we build for vertical compound engines for direct service work, where more uniformity of flow is desired than can be readily secured by two single acting plungers. In order, however, to obtain any advantage in the flow of water from differential plunger pumps, it is necessary to use at least a two-crank engine, and set the cranks at 90°, and I cannot see that the use of differential plunger pumps with single crank engines accomplishes any useful result, but on the contrary it compels the

construction of pumps requiring two sets of stuffing boxes or an inside sleeve, which is sure to leak. For this reason we use outside packed plungers wherever possible, thus securing the certain detection of leakage, and only use inside plungers or pistons on engines of small size working against low heads, and pumping water entirely free from grit.

The experience with the No. 3 engine at Chestnut Hill, Boston, shows what may be expected of unpacked inside plungers, even when working under very favorable conditions.

PUMP VALVES.

Our earlier engines were fitted with double seated brass valves of the so-called "Cornish" type, but in 1884 we fitted a pump with small rubber valves mounted on cages, which although designed originally only for pumping the very sandy water of the Missouri river under low head, were so successful as to lead us to adopt them as our standard type, to the exclusion of all others. The so-called cages are light octagonal or hexagonal iron castings, resembling very closely a hexagon nut. On the sides are mounted small rubber valves ranging in size from 2½-inch to 4-inch diameter, the sizes in most common use being 3-inch and 3½-inch diameter. In addition to the valves mounted on the sides of the cages, there are usually three or four on the top of the cage, each cage carrying from 15 to 35 valves, according to the size of the pump. There are usually 7 suction and 7 discharge cages for each pump plunger, and occasionally as many as 12 cages for very large pumps, or as low as three cages for small machines. Each cage is held in place by a single bolt extending downward through the center, tapping into a rib in the pump casting. As the pressure is always downward on these cages there is practically no strain on this bolt, and in a number of cases where the bolts have been practically twisted off by unnecessary strain when putting them in, the cages have worked for months, although the bolts were broken, being held to their seats only by the stickiness of the rubber gasket. The removal of this single bolt permits the entire cage to be lifted out for the inspection or removal of the valves, although they can be both inspected and renewed with the cage in position. The fact that nearly all of the valves are working on edge has sometimes been raised as an objection, but as the rubber valves are of

nearly the same specific gravity as water, their weight as far as any wear is concerned, may be entirely neglected, and in practice it is found that the valves on the sides of the cages wear as little as those on the top.

There can be no general statement as to the durability of these valves, much depending upon whether the water carries sand, and also on the composition of the rubber used, as this should be varied with every considerable variation in water pressure. I have in mind, however, two pumping engines, both of 18,000,000 gallons capacity, one working under 310 feet head and the other under 155 feet head, which have had the same set of rubber valves in use for over seven years, and the discharge valves are still so tight as to permit men to work beneath the delivery valves with the full water pressure on top of them.

For valve area we usually supply about 100 per cent. of the plunger area as the minimum, and running up to 125 or 150 per cent. for plunger speeds of 200 to 250 feet per minute.

It can hardly be said that there is any fixed speed proper for pumps, as different conditions demand different treatment, and while we have built pumps which are running successfully at 450 feet per minute plunger speed, and have several running entirely satisfactorily at 300 feet per minute, yet our experience leads us to advocate speeds of about 200 to 225 feet per minute for ordinary water works conditions. While high speed is quite possible, it is of more advantage to the builder of the engine than to the owner thereof. High piston speed adds but slightly, if any, to the steam economy, and usually adds considerably to the cost of maintenance of the engine.

The mechanically closed Riedler pump valve, which promised so much a few years ago, seems to have made but little progress in water works engines, at least in America. The few machines which have been built were mostly for underground work in mines, where the high speed engine was desirable on account of the small space required, but even in this field the automatic valve pump seems to be more satisfactory, as in some instances new engines of the automatic valve type are being placed in mines having previously used the mechanically controlled valve. In some cases the closing mechanism has been removed from these valves and the pumps found to work equally well without it, which leads to the

conclusion that the quiet working of these pumps at high speeds is due principally to the low lift of the valve. This brings us to the general proposition that moderately high speeds and smooth working can only be obtained by using valves of low lift, (and if of common construction, small diameter). The speed in feet per minute is not of itself of more importance than the number of strokes per minute, as the element of time plays an important part in the seating of a valve. The larger the diameter of the valve, the consequent greater lift required, and longer time for seating necessary, and the old idea of 100 to 120 feet per minute was largely due to the fact that the direct acting steam pumps used valves of comparatively large diameter, and their piston speed was maintained to the end of the stroke (instead of slowing at the end of the stroke, as with the crank engine), causing the valves to seat with a slam if operated at speeds much above 100 feet per minute.

In connection with the subject of plunger speed, the indicator diagrams shown in Fig. 10 may be of interest. No. 4 is from an Allis triple expansion engine; No. 5 from an Allis horizontal compound engine; No. 6 from Riedler valve pump (at Chestnut Hill), and shows nothing to especially recommend the mechanically closed valve. I do not consider, however, that indicator diagrams from pumps tell the whole story, for I have seen very "ragged" cards from pumps which, according to basin measurements, were working with less than 1 per cent. loss of action, while on the other hand, card No. 7 (from Louisville engine), which is practically perfect, was taken from the pump during a test on which the experts reported a loss of action averaging about 6¾ per cent. Our own experience with rubber valves of small diameter and low lift, working under suction lifts not exceeding 10 feet, is that the loss of action ranges between ½ and ¾ of 1 per cent. This is based on both weir and basin measurements.

CONDENSING APPARATUS.

Our pumping engines are fitted with jet condensers, unless bad water or lack of water for condensing purposes renders the use of surface condensers desirable. Jet condensers are preferable in many ways, but where circumstances demand the use of surface condensers, great care should be taken to avoid getting grease in the boilers. Air and feed pumps are always driven directly from

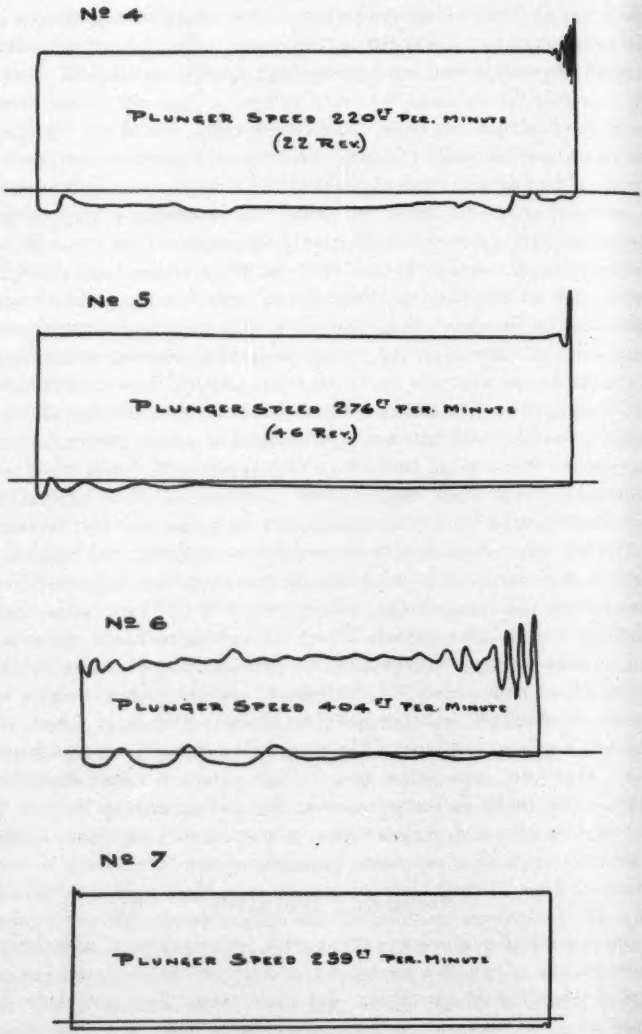


Fig. 10.
PUMP DIAGRAMS.

the main engines, and in vertical machines usually from the main pump plungers, and as there are no jacket pumps, the use of auxiliaries is entirely avoided, which, on account of the wastefulness of small steam pumps, etc., seems almost imperative if the highest results are to be attained. An illustration is seen in the No. 3 engine at Chestnut Hill, Boston. This engine, according to a test by Prof. Miller, showed the unequalled steam economy of 11.22 pounds of steam (actually supplied to the cylinders and jackets) per I. H. P. per hour, which, assuming that the boiler evaporated 10 pounds of water per pound of coal, corresponds to a duty of 160,000,000 on coal, but apparently the boiler evaporated about 10½ pounds, so that with 11.22 pounds of steam per I. H. P. per hour, and 10½ pounds of evaporation, the duty of this engine should have been about 168,000,000, while the duty on coal as reported was only about 150,000,000. In other words, the auxiliaries necessary for the operation of the engine, reduced the duty from 168,000,000 to 150,000,000, a loss of over 10 per cent.

BOILERS.

Not being interested either in the design or manufacture of boilers, my opinion as to the various types may be considered non-partisan. I do not believe that there is more than 10 per cent. difference in efficiency between any of the recognized standard types of boilers now in the market. Taking the ordinary horizontal tubular at 100, the best internally fired boiler would rate 105, and brick set water tube boilers at 95. The inferior efficiency of the horizontal tubular and water tube boilers, as compared with the internally fired boiler, is, I believe, almost entirely due to the radiation and air leakage in the brick boiler settings. There being so little difference in the efficiency of the boilers, the question becomes largely one of first cost, and details of mechanical design and construction. Where steam pressures of not over 125 pounds are to be carried, the ordinary horizontal tubular boiler answers every purpose, and all things considered, it is perhaps the most generally satisfactory. Higher steam pressures, however, render the adoption of water tube boilers, or the more expensive internally fired boilers, desirable. Internally fired boilers, to my mind, should be of the cylindrical type to avoid flat staybolted surfaces.

STEAM PRESSURES.

Marine practice has set the pace for higher steam pressures, due largely to the effort to produce the maximum H. P. with the smallest possible engine, and while high steam is undoubtedly desirable, it must be remembered that stationary engines are not subject to the same requirements as marine. The tendency, however, is rightly towards higher pressures, but for practical reasons I do not at present advocate the use of pressures much above 150 or 160 pounds, although as far as any economy is concerned, 200 or 250 pounds would be better. At 150 pounds, ordinary piping, valves, non-conductors, etc., can be used and no expensive or unusual construction is incurred, either in the boiler plant or engine, nor is special cylinder lubrication necessary. At higher pressures, however, everything becomes special, and the very slight increase in steam economy hardly warrants the extra expense and care incurred.

The question as to what steam pressure to carry, is one to be determined largely by the type of boiler selected. If ordinary horizontal tubular boilers are to be used, the steam pressure should not be over 125 pounds, although in extreme cases it is carried as high as 150 pounds. For pressures above 125 pounds, any of the various types of water tube boilers may be used, and if one is not limited as to the first cost of the plant, slightly higher efficiency may be obtained by using especially designed internally fired boilers. There is an erroneous impression that there is nothing to be gained by using triple expansion engines with steam pressures of less than 160 pounds. Our experience has shown, however, that with steam at any pressure above 80 pounds, the triple expansion engine will show an economy of from 10 to 15 per cent. better than a compound engine of the same general construction, using steam at the same pressure, and 75 per cent. of the triple expansion engines which we have built are operated with steam pressure of 125 pounds or less, the lowest being 80 pounds.

DUTY.

Next to reliability, the question of duty most interests the managers of water works.

While the coal duty in everyday service is the true test of efficiency of a pumping plant, (and also of the efficiency of its man-

agement), so many of the elements which go to make up this efficiency are entirely beyond control of the engine builder that it is hardly fair to make official duty trials on a coal basis, particularly as coal tests of less than 24 hours duration are practically valueless. Few engine builders are makers of boilers, and if they are the question of quality of coal, efficiency of firemen, draft of chimney, etc., so affect results as to render coal guarantees largely guess work. On the other hand, engine design has gotten to be a reasonably exact science, and builders can estimate the steam economy of any type of engine very closely. Therefore, it seems proper to make official duty trials on the basis of steam consumed by the engines, or on the heat unit basis recommended by the American Society of Mechanical Engineers. The work done should, in the case of outside packed plunger pumps, be figured on the basis of plunger displacement, after making proper test for leakage of valves, for the reason that with packed plungers the pump itself is a more accurate meter than a weir or Venturi meter.

The contention is frequently made that high duty engines fall very much below their test records on everyday service, and while in the majority of cases it is to be expected that the average duty would be somewhat less than that shown on a test, owing to the difficulty of watching all small matters on a continuous run which are looked after very sharply on a short test. But the greater part of the falling off in duty under everyday conditions is due to the fact that the engines are not run under contract conditions, either as to speed of the engine or steam and water pressures. The first is of much less importance than the last, for our experience has shown that a high duty engine running at one-half its rated capacity will give within 10 per cent. of as high economy as it will running at full capacity, provided steam and water pressures are kept the same. It is a very common, practice, however, when purchasing a pumping engine, to specify the maximum requirements as to water pressure, and the duty test is made under these conditions, while in regular service the water pressure may be 10 to 25 per cent. less than that originally specified. This means that in regular operation steam is carried at a considerably lower pressure, and the engine is throughout very much underloaded, and therefore necessarily uneconomical.

A properly designed pumping engine will work with water pres-

sures from 10 to 25 per cent. higher than the nominal pressures, without appreciably falling off in economy, but if the water pressures are from 10 to 25 per cent. *less* than the engine was designed for, the falling off in economy becomes very marked. It will, therefore, be seen that it is very important that specifications for pumping engines should call for the duty tests to be made under ordinary working pressures, the engines being guaranteed to work properly under maximum conditions. It is also desirable that the engine should be built as nearly as possible of the capacity required for ordinary work, for while not as important as the matter of water pressures, still, if the best results are to be obtained, it is desirable to have the engine run at approximately its rated speed. Of course on direct service work the speed of the engine must necessarily be subject to considerable variation between the hours of maximum and minimum consumption.

The efficiency of a boiler plant is very frequently not maintained to its full standard. More boilers are often operated than are actually required, thus burning less coal per square foot of grate than is economical, and there are numberless opportunities about the plant for wastes, which, while individually small, in the aggregate reach a very appreciable item.

In a general way, therefore, the falling off in economy under ordinary running conditions, is due not to faulty construction of the machinery, but the fact that it is operated under different conditions than those named in the specification, the requirements of which it was designed to meet. Therefore, the matter rests more in the hands of the purchasers of the machinery than in those of the builders.

It is unfortunate that the keeping of accurate records at all pumping stations is so rare, and the recommendation of your Association on this subject is to be commended.

What a modern high duty engine will do under regular running conditions is well shown by the record of an 18,000,000 gallon triple expansion engine in the Milwaukee Water Works, (See Fig. 11). This engine has been in service a little over seven years, averaging over 23 hours per day at full speed. It pumps about 70 per cent. of all the water used by the city, the remaining 30 per cent. being pumped by compound engines, the newest of which is 17 years old. The duty of the station on all coal burned for all

Fig. 11.

YEARLY RECORD OF THE MILWAUKEE 18,000,000 GALLON TRIPLE EXPANSION PUMPING ENGINE.

Started Aug. 23, (new).....	1891.	1892.	1893.	1894.	1895.	1896.	1897.	1898.
Total hours run per year.....	1914	6979½	8287½	8117½	8183¼	8737½	8005	8666¼
Average hours run per day.....	$14\frac{7}{10}$	19	$22\frac{45}{100}$	$22\frac{35}{100}$	$22\frac{42}{100}$	$23\frac{87}{100}$	22	$23\frac{74}{100}$
Average gals. pumped per day for 365 days.....	10,870,000	14,250,000	16,700,000	16,500,000	17,830,000	19,750,000	16,360,000	18,445,000
Rate of pumping, Actual running time.....	17,750,000	18,000,000	17,850,000	17,800,000	19,100,000	19,850,000	18,000,000	18,650,000
Per cent. of total pumpage of station for year.....	16%	63%	65%	64%	$70\frac{7}{10}$%	$79\frac{8}{10}$%	$69\frac{3}{10}$%	79½%
Duty of station on all coal burned	80,428,800	99,043,000	101,481,000	103,342,000	103,165,000	104,552,000	105,125,000	109,172,618

Total cost to pump 1,000,000 gallons to reservoir (157½ foot head) for years 1897 and 1898, $3.88.

Total cost to pump 1,000,000 gallons one foot high for the years 1897 and 1898, $2\frac{47}{100}$ cents.

These costs cover all expenses for all engines at the station, including wages, coal, oil, repairs and supplies.

The effect of the higher duty of 1898 does not reduce the cost of pumping below that for 1897; for the reason that the price per ton for coal was higher in 1898 than in 1897, and further, the quantity pumped in 1898 was 2½% less than in 1897, while the wages and other expenses remained the same.

Duty is based on all coal burned for all purposes including heating building, running shop engine during day, and electric light at night. No deductions for ashes.

Previous to August, 1891, all of the water was pumped by *compound* engines, and the annual duty of the station on good anthracite coal was about 80,000,000. The *triple* usually pumps about 70% of the water, most of the balance being pumped by a *compound* built in 1881. The duty of the station has risen about 35% since the introduction of the *triple*, and if all the water could be pumped by *triples*, the duty would probably be 50% higher than it was when the work was done by *compound* engines.

purposes, without deductions or allowances of any kind, in 1891 was about 80,000,000, all of the water being pumped by compound engines. The first year that the triple ran (pumping only 63 per cent. of the water) the duty rose to 99,000,000, and has since steadily risen until the year 1898, when it was 109,172,618 foot pounds per 100 pounds of coal burned, the coal being Youghiogheny slack, costing $1.78 a ton, delivered at the station, and carrying 14 per cent. ash. While this engine ordinarily takes steam from the same boilers which supply the compound engines, it recently became possible to separate it for a period of five weeks, (Fig. 12) and during that time the engine averaged $23\frac{72}{100}$ hours per day, running at full speed, and gave a duty of of above 122,000,000 foot-pounds per 100 pounds of coal. The cost of coal to raise 1,000,000 gallons $157\frac{43}{100}$ feet, was $95\frac{1}{2}$ cents, the coal per indicated horse power per hour being less than $1\frac{1}{2}$ pounds.

Fig. 12.

MILWAUKEE 18,000,000 GALLON PUMPING ENGINE.

Record for Five Weeks, July 31st to Sept. 3d, 1898.

Total days run	35
Total hours run	$830\frac{1}{4}$
Average daily run per day	$23\frac{72}{100}$
Total gallons pumped	632,893,750
Average gallons pumped per day	18,075,000
Average head pumped against	$157\frac{43}{100}$ feet
Total coal burned	679,500 pounds
Duty on total coal	122,291,111 foot pounds
Coal—Youghiogheny slack:	
Cost of coal per ton	$1.78 delivered at station
Total ash and clinker	95,096 pounds
Per cent. ash and clinker	14%
Pounds of coal to raise 1,000,000 gallons 157.43 feet high	1,074 pounds
Cost of coal to raise 1,000,000 gallons 157.43 feet	$95\frac{1}{2}$ cents
Pounds of coal to raise 1,000,000 gallons one foot high	$6\frac{82}{100}$ pounds
Cost of coal to raise 1,000,000 gallons one foot high	$\frac{6}{10}$ of 1 cent
Indicated H. P. of engine	540 (approximate)
Coal per I. H. P. per hour	$1\frac{1}{4}$ pounds
Cost of coal per I. H. P. per year running 10 hours per day, (5,480 lbs.)	$4.87
Cost of coal per I. H. P. per year running 24 hours per day, (13,152 lbs)	$11.69

When it is considered that this record is made by an engine which has been in continuous operation for seven years, without having the cylinder heads removed, or even the steam valves removed for examination, the result is certainly very good.

The yearly performance of the 5,000,000 triple at Brookline, Mass., given below, is also an example of what may be done with a high duty engine, although the fact that the Brookline engine is only operated about eight hours daily, is of course fatal to the best results on the basis of all coal burned during the year.

RECORD OF WORK DONE AT THE LOW SERVICE PUMPING STATION OF BROOKLINE, MASS.

Year.	Gallons pumped.	Coal consumed in pounds for all purposes. No deductions.	Gallons per pound of coal.	
1890 ..	322,767,796	1,141,175	282.8	Direct acting low duty engine.
1891 ..	356,812,572	1,263,395	282.4	
1892....	390,593,090	1,351,032	289.1	
1893	443,973,099	1,582,471	280.5	
1894....	481,633,366	998,107	482.5	Triple expansion engine, in use part of the time.
1895....	483,969,131	803,642	602.2	In use most of the time.
1896	494,095,562	745,692	662.6	In use all the time.
1897....	508,174,048	761,826	670.0	In use all the time.

Average number of gallons pumped per day in 1897.....	1,392,258
Coal required if old engine had performed this work.....	1,791,238 lbs
Coal actually consumed.....	761,826 lbs
Coal saved.....	1,029,412 lbs

Under test conditions compound engines with steam from 100 to 125 pounds, 110,000,000 to 115,000,000 duty per 1,000 pounds of steam, is all than can be expected, although in a few cases where very high steam has been carried, or other conditions are particularly favorable, this duty has been very considerably exceeded, notably in the case of Mr. Leavitt's engine at Louisville and the four Allis compound engines at Pittsburg, (Fig. 6).

For triple expansion engines, with steam at from 90 to 110 pounds, duties of 120,000,000 to 130,000,000 may be easily attained, while with steam pressures at from 125 to 150 pounds, a triple expansion engine will easily run up to 140,000,000 to 150,000,000, and very probably with steam pressures from 175 to 200 pounds, under favorable conditions the triple will give 160,000,000 duty per 1,000 pounds of steam.

Just what the future will show in the direction of duty it is difficult to predict. The quadruple expansion engine using high pres-

sure steam would undoubtedly effect some increase in economy, but on account of the satisfactory mechanical construction of the present triple expansion engine, I consider the adoption of the quadruple as doubtful, until our steam pressures exceed 200 pounds.

Our present ideal is to produce a horse power on 10 pounds of steam, and if our boiler designers will give us an actual evaporation of 10 pounds of water per pound of coal, we shall have a horse power with one pound of coal, or approximately 180,000,000 duty on coal. But the case is something like the race for speed in steamship practice, and we have approached a very steep part of the ascending curve of efficiency, so that the last fraction of duty comes very hard.

To sum up, the increase in duty during the past 25 years has been due principally to the introduction of the compound engine and also the triple expansion engine, both carrying higher steam pressures than were general in former years. A considerable part of the improvement is due, not to any radical change in the general construction of steam engines, but to the more careful design of the many details which go to make up a successful machine.

The reduction in cost of pumping machinery has only followed the reduction in cost of all classes of machinery, and in fact nearly all classes of manufactured products, due to improved facilities for manufacture, and it does not now appear reasonable to expect any very material reduction in the cost of machinery of this class for many years to come, unless entirely different and cheaper designs are adopted.

DISCUSSION.

THE PRESIDENT. The paper is now open for discussion.

MR. DEAN. I am not going to discuss the engine portion of Mr. Reynolds' paper, but wish to the boiler part of it. I think it is a great mistake to perpetuate the idea, which I consider a fallacy, that when you wish to use high pressure you must have a water tube boiler. What high pressure is, seems to be an uncertain thing, because some people say 125 pounds is high pressure, and some people say something else is. As for my own feeling, I think 125 pounds is pretty low. Now, I take the position that no man can give any reason why a horizontal return tubular boiler should be limited to 125 pounds, or even 150 pounds.

To show that it is not a question of insurance, I will say that a short time ago I wrote to the Hartford Steam Boiler Insurance Company, (and the letter was answered by the president), asking them if they would insure a 72-inch horizontal return tubular boiler for 160 pounds of steam, provided it was made of ½-inch plates with drilled holes and butt longitudinal seams having inside and outside covering plates, and having a number of rows of rivets which would not go through both of the covering plates. The president wrote to me that if the boiler was made in this way with drilled holes and everything else as good as could be, they would insure the boiler for 160 pounds of steam. A few days ago I asked the engineer of another of the most prominent boiler insurance companies whether or not they had any notions or superstitions. He said he hoped not. Said I: "Do you object to insuring an 84-inch horizontal return tubular boiler with ½ inch plates with a factor of safety of five referred to the ultimate strength." After a little talk he said, "No."

In 1890 there was read before the Institution of Engineers and Shipbuilders of England, (East Coast), by one man named Kilvington and another named Taylor, jointly, a paper in which they said that they had furnaces in marine boilers which were over ⅝ of an inch thick, that had been in use for eight or ten years, and nothing had happened to them; that they had had furnaces in use for three years ¾ of an inch thick, carrying 160 pounds of steam, to which nothing had happened so far; and that they would not hesitate, if there were any occasion, to use furnaces 1 inch thick.

Now, we know that from theoretical considerations there isn't any reason why 1-inch plates are not just as safe as thinner ones, for the reason that the heat passes through a plate with tremendous facility; and the trouble is (if there is any trouble) to get the heat into the plate and to get it out. Of course, when there is oil on the water side of a plate, there is likely to be trouble. This trouble would be just exactly the same whether the furnace is 1 inch thick or less, for if it explodes it will do just the same thing. It is a good deal like the danger in carrying a high pressure; if a boiler explodes with a low pressure it will kill everybody around it and smash everything all to pieces, and that it is just what it will do in the other case. I think it is simply a question in boilers of the strain per square inch on the weakest part, and corrosion. One

very important thing about a boiler is to have it so made that you can inspect as much of the inside as possible, as well as the outside, and you can thereby see what is going on, and keep yourself safe.

There are some water tube boilers of 300 horse power units, that have 36-inch drums. These drums are just as likely to explode as a 36-inch boiler, and if they should explode they would do just the same damage.

So I fail to see why anybody should attempt to scare the public by saying that horizontal return tubular boilers are dangerous, when you get up to a higher pressure than 125 pounds. If you have a notion that half an inch plate is the thickest that you want to use, and you want to use a horizontal return tubular boiler, make your boiler so small that the strain per square inch will be what you think it ought to be. And there is a good deal in thinking what it ought to be rather than knowing what it ought to be. A factor of safety referred to the ultimate strength I do not think of any consequence; referred to the elastic limit, it is of some consequence. I think a factor of safety of two, referred to the elastic limit, would be enough, but of course none of us have quite the courage to use so small a factor, especially when we consider corrosion. I will call attention to the fact in connection with marine furnaces, that a few years ago, it was thought $\frac{5}{8}$ of an inch was the greatest thickness which could safely be used; but those engineers of whom I have spoken have gone beyond that without any disastrous results.

MR. SMITH. Mr. President, there has been one point brought up which I must say I cannot thoroughly agree with, and that is on the matter of using a single acting plunger on a vertical pump in contradistinction to using a double acting plunger on a vertical pump. As I understood the paper the reason assigned for that was that the flow of water would be greater at the bottom of the pump and less at the top. That, to my mind, is a simple matter, which can be easily overcome by a proper design in the vacuum chamber on the suction, and by a proper design of the valve area between the two pump tables, the two valve tables. I have in my pocket a card, which I would like to pass around, taken from a double acting vertical pump, from the top side, in fact I have one taken from both the top and bottom sides, and I fail to understand the force of

the statement which has been made. These cards were taken from a double acting vertical pump working against a pressure of 145 pounds.

MR. REYNOLDS. In reply to Mr. Smith, I want to say that in this paper I referred particularly to very long stroke pumps working under high suction lift. Let us imagine a case where the suction lift to the bottom valves was 28 feet; how would the top of your pump fill on a 9-foot stroke engine?

MR. SMITH. I would say in reply to that, that a man who set a pump in any such manner didn't fulfill the conditions necessary in setting such a pump.

MR. REYNOLDS. While the case as stated may be an extreme one, it is often approached in actual practice on western rivers, where the rise and fall amounts to from 50 to 70 feet. In such cases it is necessary to set the pumps in a pit at such depth as to place the suction valves within reach of extreme low water in the river. Many pump pits along the rivers are over 40 feet deep, those now under construction in Cincinnati being 85 feet deep, and the pits at St. Louis are 55 feet deep. With engines of the long strokes employed in some places, (6 to 9 feet) the pits would require to be 8 to 10 feet deeper if double acting pumps were used, and this increased depth of pit would mean a serious expense without any gain whatever as far as the action of the pumps is concerned, but on the contrary there would still be a difference of several feet in the suction lift between the top and bottom valves which would make an appreciable difference in the behavior of the two ends of the pump. The cards which Mr. Smith shows are excellent, and as his engine is one of comparatively short stroke, I believe, and doubtless working under very moderate suction lift, there is no reason why both ends of the pump should not fill perfectly, though with a single crank vertical engine of the type I understand he has, I fail to see any particular advantage in making the plungers double acting.

I think as I said before, Mr. Smith failed to notice that my original statement as to the objections to double acting vertical pumps, applied particularly to long stroke machines with high suction lift, and not to short stroke pumps under ordinary lifts, against which there is no particular objection.

MR. DEAN. Going back to the horizontal return tubular boilers, I made a statement recently before a number of mill men, that I did not think any such boiler with butt joints had ever exploded. I wrote to the president of the Hartford Steam Boiler Insurance Company asking him if he had ever known of such an explosion, and he replied that he never had. All these horizontal return tubular boilers that explode are made with lap joints, and they explode at the seam. Now, the Hartford Steam Boiler Insurance Company certainly knows about boilers that explode, how many explode, and all about them; and the president says so far as he knows, no boiler of that type which has been built with butt joints, has ever exploded. This should be a very impressive fact.

MR. REYNOLDS. In reply to what Mr. Dean has said, I would say that I am not interested in advocating any particular type of boiler, and I think the inference that my paper is an "attempt to scare the public," on the horizontal tubular boiler question, for the benefit of the water tube boiler, is unwarranted. My own practice has been to use horizontal tubular boilers for all pressures up to 125 pounds, but in the west there is a strong feeling (I agree it may be largely a superstition, as Mr. Dean intimates), against using plates thicker than about $\frac{1}{2}$ inch on this type of boiler. Perhaps the fact that most water in the west carries a considerable amount of lime and not infrequently mud, may have something to do with this feeling.

The vertical fire tube boiler which has been so successful in New England, has never gained any foothold in the west, owing very largely to the deposits of scale and sediment on the bottom tube sheets, although in cases where properly cared for, the vertical fire tube boiler has done well. Of course throughout New England the water is usually so good that the question of scale or deposit may be almost entirely neglected, but practice in different parts of the country naturally varies somewhat. But as a designer of pumping engines I shall not be particular as to the type of boiler which is used in connection with our engines, so long as it gives the highest results, and if our boiler makers or designers will guarantee the efficiency which Mr. Dean states was readily reached in New Bedford, we shall then get a duty on coal which both boiler and engine man may well be proud of.

Mr. Rockwood's name being mentioned, the president called upon him.

MR. ROCKWOOD. I am a guest, Mr. President, and it would be hardly wise for me to precipitate a compound engine struggle here. I was very much interested in the paper. I like to have a man say flatly what he believes, aud then you know just where you disagree with him. I wish with all my heart that the big Allis pumping engine, now to be seen running at the Chestnut Hill reservoir pumping station, could be tried without its intermediate steam end, for it is my belief it wouldn't show any difference in the coal bill at all.

MR. REYNOLDS. In answer to Mr. Rockwood, I would say that the opinions expressed in this paper are entirely personal ones, and are not necessarily accepted by the public. I think the same applies to what Mr. Rockwood has just said. (Laughter.)

THE APPLICATION OF GAS, GASOLINE AND OIL ENGINES TO PUMPING MACHINERY.

BY FREEMAN C. COFFIN, C. E., BOSTON, MASS.

[*Read Dec. 14, 1898.*]

During the past two years the use of gas, gasoline and oil engines in driving pumping machinery has been increasing rapidly. The adoption of this class of power in the Western or Middle Western states has preceded its application to pumping machinery in New England, especially for public water supplies. There are already a large number of public pumping plants in the former section driven by some form of internal combustion engine, while in New England there are as yet very few such plants. This may be largely due to two causes; one the lack of gravity systems by which so many of the small towns and villages of New England are supplied; the other the presence of natural gas in the former section which provides a cheap fuel for this type of engine.

It is evident that under present conditions and in the present status of steam power there is a field for the application of this type of engine to pumping water for places of moderate size. Wherever the amount of water to be pumped is such that some type of engine using steam expansively, or the so-called "high duty pumping machinery" is more economical than the compound condensing direct acting pumping engine, the internal combustion engine now actually on the market is no competitor.

Wherever the compound condensing direct acting engine which has given and is giving such admirable results in places of moderate size, would be used, it is perhaps an open question whether to adopt that type or a gasoline or oil engine and power pump, except in cases where a steam boiler plant is already established and in operation, then steam pumping machinery would be the most economical. For very small plants the internal combustion engine has advantages that in many cases must cause its adoption where a comparison is carefully made.

There is apparently no question of the feasibility of using this type of power for pumping. It is not in the experimental stage but it is settled by experience that it is safe and certain in operation if an engine of good design and construction is used.

It is not the purpose of this paper to discuss the theory of the internal combustion engine or to enter into a lengthy description of it. Those wishing to consult a treatise on these engines are referred to "The Gas and Oil Engine" by Dugald Clerk.

A few of the general characteristics of these engines will be briefly mentioned.

The gas and gasoline engines are practically alike in construction and operation, the same engine can be used with either fuel, a few alterations being required in the arrangement when the fuel is changed. The oil engine is similar in general appearance and operation but the treatment of the fuel in the engine differs from that of gasoline.

In a gas engine the gas is introduced into the cylinder and mixed with air which is drawn through a valve into the cylinder by the outward stroke of the piston, the mixture is compressed by the return stroke and fired by an electric spark, or an ignition tube.

The cylinder is open on one end. The explosion of the air and gas behind the piston drives it forward and imparts the energy to the fly wheel, which is very heavy. There is but one explosion in two complete revolutions, the return of the piston forcing the gases formed by the combustion out at the exhaust, the next forward stroke drawing in the air and admitting the gas and the return stroke completing the cycle of work and compressing the charge for the next explosion. The gasoline engine works in precisely the same way except that the fuel is forced in by a pump worked by the revolution of the engine and is turned to gas within the cylinder or combustion chamber before mixing with the air.

The principal difference in the working of the oil engine is, that the fuel not being as volatile as gasoline, is introduced in a fine spray to a vaporizer where it is turned into vapor by the heated walls and then mixed with air. The vaporizer must be heated by a special lamp before starting the engine; this requires from seven to ten minutes, and therefore the oil engine requires that much more time to start it, than the gas or gasoline engine, which simply requires the supply cock to be opened, a few turns to be given to the

fly wheel and it is off, if there is no difficulty with the battery which provides the spark for firing. This seems to be the weak point in the gas engine, at least the only difficulty that I have seen in starting and running them seems to be connected in some way with the battery. Some of the oil engines require no battery, the charge is ignited after the engine is started by the heat of the walls of the vaporizer in combination with the pressure produced by the return of the piston. I can not see that the gasoline is superior to the oil engine nor vice versa, each have their points of superiority and each have slight defects. The cost of fuel is probably slightly less in the gasoline engine, while the use and storage of kerosene stands higher in popular favor. All engine cylinders are jacketed and a stream of cold water is passed around them, about 10 gallons per H. P. per hour are used with the oil engine.

The waste gases are exhausted into the air and if the exhaust pipe is carried above the building very little odor is noticeable. The exhaust is noisy unless muffled in some way. The speed of these engines is from 200 to 400 revolutions per minute, according to the size, and the main shaft can be connected directly to the pinion shaft of a power pump. It is necessary to use a friction clutch for this connection, as this type of engine must be started light and the load applied after it has acquired its speed.

ATTENDANCE.

Perhaps the most important feature in the operation of internal combustion engines is the attendance. Anyone with ordinary intelligence and no training as an engineer, can be taught in a short time to run one. In a well designed plant, properly supplied with large oil cups, the necessary attendance is limited to starting the engine, providing a sufficient supply of fuel in the tank and oil in the cups and stopping it at the proper time. Starting under ordinary conditions requires from a minimum of one minute with gas or gasoline engines to a maximum of fifteen with oil engines.

The speed of these engines seems to be most perfectly regulated by the governor which controls the supply of fuel, a sudden variation in load making but slight change in speed. If a main should break in front of the pump the speed would hardly vary 10 per cent. On the other hand, there is very little adjustment of speed possible. About 15 per cent. from the rated speed either

way is as much variation as can be obtained. If a pumping plant were required in which the running capacity could be reduced 50 per cent., it would be necessary to use two small pumps, one each side of the engine, with the friction clutches, either or both pumps could be run. As the engine so readily adapts itself to a change of load a pump could be thrown on or off at any time. These engines run very economically with a small load.

In writing the foregoing I have described the engines as I have found them in my own experience, and through the investigations that I made before recommending their use.

Very recently, however, a pamphlet has come into my hands, written by a Western man, presumably an engineer, although it is not so stated, in which a very strongly adverse opinion of internal combustion engines is expressed and a severe attack made upon oil engines through a certain make of that type. The writer of this paper has no acquaintance with the make of engine thus criticized.

It would seem from a reading of the pamphlet that no engines of that kind have ever been run successfully and the conclusion is drawn, that no internal combustion engine can be successful. This conclusion seems to be contrary to the general experience, as I have found it among the users of these engines. Occasional difficulties will no doubt be met, but with a first-class engine of sufficient capacity for its work and properly erected I believe its performance will generally be satisfactory.

Certainly no plant of this kind should be installed without a careful study of all of the conditions and a comparative study of a steam plant for the same service.

CONDITIONS OF PUMPING UNDER WHICH INTERNAL COMBUSTION ENGINES OFFER ESPECIAL ADVANTAGES.

Wherever it is desirable to have a pumping plant run by the superintendent of the works, or by some one who has other duties to perform, the small amount of time required for attendance and the fact that a skilled or licensed engineer is not required tells strongly in favor of these engines when compared with steam.

Where a needed additional supply of water can be secured at some place too remote from the existing pumping station to be pumped by that plant, the smaller first cost and possibility of running the new plant without occupying the full time of an engineer

would naturally lead to the adoption of this type of engine. Where an extra supply is needed for a few months in the year a pumping plant can be installed at a low cost that will do satisfactory and economical work at less annual expenditure for fixed charges.

When a suitable and sufficient supply can only be obtained by developing two or more sources as large springs or limited driven well supplies, a plant of this kind can be installed at each place at moderate cost and run by the same person.

High service supplies where the high service pumping cannot be done at the main station or where the low service is supplied by gravity.

A pumping plant of this type seems peculiarly adapted for use in a sewerage system where the topography requires that the sewage be collected and pumped at two or more points in order to avoid great expense of construction. As the lift is generally light the total power required is small and even in fuel consumption these engines would compare favorably with a very good type of steam engine.

I believe that its merits demand an economic comparison with steam power in any plant where the amount pumped does not require a high duty engine.

Every case must be decided upon its own merits and I am far from believing that these engines will supplant steam or from recommending them in any sweeping manner, however desirable they may be under certain conditions.

During the past season the writer has installed two pumping plants driven by oil engines. The oil engine was chosen in both cases, not on account of any apparent superiority over the gasoline engine, but because the parties interested preferred the storage and use of kerosene to that of gasoline, although realizing that the cost of fuel would be somewhat more with the former.

The cost of a first class engine of either type when installed with all appurtenances did not differ materially at that time and for those sizes.

COHASSET WATER WORKS PLANT.

The first of the above mentioned plants to be installed was for the Cohasset Water Company. This plant was to pump a small additional supply from driven wells. The available supply was

thought to be sufficient, in addition to the original one, for several years, but not large enough to warrant much outlay, therefore the plant was put in with as little expense as possible. A vertical triplex single acting pump was used. The plungers were 6½ inches in diameter by 8-inch stroke with a capacity of 150 gallons per minute at its normal rate of speed. The pump was of an inexpensive type with outside bearings on the main shaft and plungers of the trunk pattern, with no cross-heads and the connecting rod hinged to the bottom of the plungers. It has, however, given very satisfactory results running without attendance throughout the season. It was built and erected by the Geo. F. Blake M'f'g. Co.

The engine was a Hornsby-Akroyd oil engine rated at 13 H. P., built and erected by the De La Vergue Refrigerating Co. The engine shaft was directly connected to the pinion shaft of the pump by a friction clutch.

ACCEPTANCE TEST.

The engine was guaranteed to develop at least 13 H. P. with a consumption of 1 pound of oil (150° test) per H. P. per hour. The pump was guaranteed to give 70 per cent. efficiency. The engine was first tested alone by an absorption brake, the test resulting in the development of nearly 15 H. P. with a consumption of 0.926 pounds of oil per H. P. per hour. The combined plant was then tested under as nearly as possible the same condition as in the brake test, the water pumped and the total lift being taken as the measure of the work. This test resulted in a consumption of 1.323 pounds of oil per H. P. per hour, based upon the foot pounds of work done in actually raising water. Assuming 1 pound of oil per H. P. as the performance of the engine the efficiency of the pump was nearly 76 per cent. Assuming that the engine was doing the same as in the brake test or developing one H. P. for 0.926 pounds of oil per hour the efficiency of the pump was exactly 70 per cent. It was perhaps remarkable that this should have been precisely the same as that guaranteed but I can only say that there was no juggling with the figures. The performance of the entire plant was about 8 per cent. in excess of the guarantee.

This plant was started May 31, 1898, and stopped for the season October 26th, and run during that time 115 days, pumping an

average of 103,600 gallons per day, or a total of 11,914,000 gallons. Deducting the amount of water used for cooling (estimated from measurements made during the test) or 173,000 gallons, the effective pumping was 11,741,000 gallons, the average lift was about 160 feet.

The total amount of oil purchased was 1,745 gallons, assuming that the engine consumed 1 pound per H. P. per hour the efficiency of the pump was 70 per cent. or about 8 per cent. for the entire plant, less than during the test.

The number of gallons pumped per gallon of oil was 6,728. The weight of oil sold as one gallon by the Standard Oil Co. is 6.5 pounds.

The attendance given this plant was limited to starting the engine in the morning and as the oil cups were rather small, oiling at noon and shutting down at night, through the day the building was locked up.

The cost of this plant, exclusive of building and foundation, was very nearly $1,500 erected, including gauges, connections, etc.

WINCHESTER OIL ENGINE.

This plant was installed for the high service system of the Winchester Water Works. The pump is vertical triplex single acting with 8½ x 12-inch plungers, and a capacity at its normal speed of 350 gallons per minute or 500,000 in twenty-four hours. Each plunger has a guided cross-head, there is a crank on each end of the shaft and overhung from the frame which carries the main boxes, the third crank is between the frames. It is one of the best types of vertical triplex pumps.

The engine is a 20-H. P. Hornsby-Akroyd oil engine; this power is 40 per cent. in excess of the actual work to be done under the requirements or 14 per cent. in excess of the power required to drive the pump if the latter has an efficiency of 80 per cent.

The entire contract was taken by the Goulds M'f'g. Co. whose guarantee was that the plant should be able to pump 350 gallons per minute against a total lift of 160 feet with a consumption of oil not exceeding 1 pound for each 10,750,000 foot pounds work done.

This plant was recently tested by Mr. J. W. Mackie representing the Goulds M'f'g. Co. and the writer.

The results of the test were as follows:

Duration of test		5 hours
Average head pumped against	142.6	feet
Total number of revolutions	12,007.	
Revolutions per minute	40.	
Displacement per revolution	8.843	gallons
Displacement per minute	354.	"
Total displacement in gallons	106,182.7	"
Slip past plungers (measured)	20.	"
Net amount pumped neglecting slip of valves	106,162.7	"
Weight of water	885,337.	"
Total work	126,247,598.	ft. lbs.
Average power (measured by water pumped)	12.75	H. P.
Total fuel (150° test kerosene oil)	70.69	lbs
Total fuel	10.876	gallons
Consumption of oil per H. P. per hour	1.11	lbs.
Consumption of oil per H. P. per hour	0.171	gallons
Work done per pound of oil	1,771,772.	ft. lbs.
Work done per gallon of oil	11,516,518.	ft. lbs.

The latter figure was 7 per cent. in excess of the contract requirement of 10,750,000 foot pounds for each gallon of oil consumed.

Although there was no requirement as to the relative efficiency of the engine and pump, it was understood that the engine should develop one horse power per pound of oil per hour, and that the pump should show an efficiency of not less than 80 per cent. of the power furnished it by the engine.

No brake test has yet been made, but if it is assumed that the engine consumed 1 pound of oil per H. P. per hour then the pump would have an efficiency of a little more than 90 per cent. If on the other hand an efficiency of 80 per cent. is assumed for the pump, the engine was developing one horse power with a consumption of a little less than 0.89 pounds of oil per hour. The head pumped against during the test was the ordinary working head. Another run was made, however, that showed the plant to exceed the capacity requirement of the contract which was 350 gallons per minute against a total lift of 160 feet. The plant was run at that rate against a head of 186 feet, showing available power 16 per cent. in excess of the requirement.

The amount of water used for cooling during the test was 8.2 gallons per H. P. of actual work done or about 6.8 gallons per H. P. developed by the engine.

It may be noted that the engine was developing but little over 15 H. P. during the test or about three-fourths load; the result might have been still higher if the full 20 H. P. had been developed.

The cost of this plant, exclusive of building and foundation, was about $2,900, including piping, erection, gauges, etc. Oil cost here at the present time 7¾ cents per gallon delivered in 300 gallon lots.

A SEWAGE PUMPING PLANT.

The internal combustion engine seems peculiarly adapted to the conditions of pumping in some sewerage systems and although not yet built, a brief description will be given of a plant, which has been designed by the writer for the sewerage system of the City of Charlottetown, Prince Edward Island. The sewage from the central portion of the city can be discharged by gravity; this part of the system is being built this season. There are two sections, one on each side of the city, from which the sewage must be pumped. The estimated flow from one is 100,000 gallons per day, from the other 150,000 gallons. It is impracticable to collect the sewage of both these sections at one point on account of the expensive deep cutting between. A small storage reservoir is designed for each section, with a pumping plant consisting of a gas engine and two centrifugal pumps at each station; the size of one pump to be about equal to the power of the engine, the other one-half that size. With the smaller one the pumping may go on nearly continuously, using the reservoir as a margin for either an excess or deficiency of flow of sewage, as compared with the capacity of the pump.

The larger pump will be used, if it is desired to pump at certain stages of the tide, or occasionally if the flow is greater than the capacity of the small pump. A recording device will show in the office the height of the sewage in the reservoirs. It is expected that the pumping machinery can be looked after and the pumping done by the man who inspects the sewers and attends to the ordinary flushing.

The pumping plant will be stopped automatically, when the sewage in the pump well is drawn down, by the action of a float, in shutting off the fuel supply of the engine. The attendance will consist in starting the engine and keeping the oil cups filled. Gas

will probably be used, as the price of gasoline is high in this place. Gasoline can be used, however, in the same engine by making a few changes and continuous pumping will not depend entirely upon the gas supply.

These plants will probably be installed the coming season.

COMPARATIVE COST OF PUMPING.

From the results secured in my experience it is estimated that a well designed oil engine plant will in regular duty raise 1,000,000 gallons one foot high for each gallon of oil consumed.

This allows the engine one pound of oil per H. P. per hour and the pump an average efficiency of 65 per cent. This I believe to be a conservative estimate of the results to be secured.

From tests made by others which there is not the space to quote here, it is estimated that a first class gasoline engine plant will do the same amount of work with a consumption of 0.9 gallons of fuel.

There is so much difference in the quality that it is difficult with the data available to make an estimate of the consumption of gas. A safe estimate, however, is that 130 cubic feet of good city gas will do the same work as a gallon of oil, or raise 1,000,000 gallons one foot high.

Upon this basis the fuel cost of pumping water will be as follows:

	With Oil at 9 Cents per Gal.	Gasoline at 9 Cents per Gal.	Gas at $1.00 per 1,000 Cu. Ft.
1,000,000 gals. one foot high...	9 cents	8.1 cents	13 cents

An estimate has been made with an endeavor to be as exact as it is possible to be in a general estimate, of the cost of installation of plants of different capacities of both types, steam and internal combustion engines, also cost of fuel, attendance, repairs, supplies and fixed charges. The steam plants estimated upon were of the compound condensing duplex direct acting type of pumping engines, with horizontal boiler and brick chimney, and small coal shed. The duties allowed such plants were estimated from the reports of water works, pumping similar quantities of water. The size of the plant of either type was such that the daily consumption could be pumped

in ten hours. The cost of the plants upon which the fixed charges are based includes the necessary buildings and foundations.

From these estimates diagrams have been prepared showing the annual cost of each item of expense and the total cost. The diagrams are based upon a dynamic head of 210 feet, this being an approximation to the average head in different places as given in water works statistics. The pumping is given as the amount pumped per day. While estimates of this kind must be used with caution in particular cases, they may be of assistance in the preliminary study of the subject. The following table gives total annual cost for certain rates of pumping:

TABLE GIVING COMPARATIVE ANNUAL COST OF PUMPING WITH DIFFERENT TYPES OF POWER.

Average Daily Pumping in Gallons.	Oil Engine. Oil at 9 cts.	Gasoline Engine. Gasoline at 9 cts.	Gas Engine.		Steam Pump.	
			Gas at $1.00.	Gas at $0.50.	Coal at $5.00.	Coal at $4.00.
50,000	$ 770	$ 735	$ 920	$ 675	$1,230	$1,160
100,000	1,275	1,200	1,580	1,035	1,740	1,600
200,000	2,200	2,050	2,815	1,820	2,525	2,300
300,000	3,085	2,875	4,000	2,510	3,130	2,850
400,000	3,920	3,640	5,140	3,150	3,700	3,350
500,000	4,745	4,400	6,270	3,780	4,200	3,790

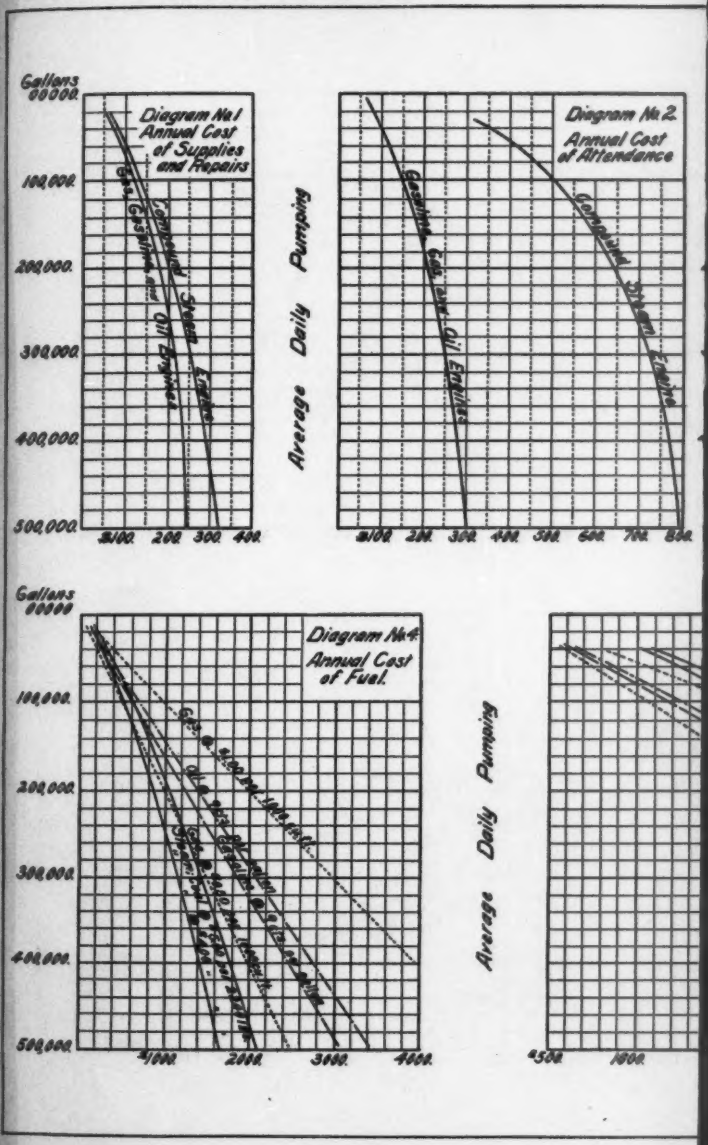
The basis upon which these comparative estimates were made was certainly not unduly favorable to the internal combustion engine.

It is important that there should be a margin of power in this type of engine over the work to be done; if they are overloaded they will gradually slow down and stop entirely.

It is also necessary that a very rigid foundation should be provided, as the shock of the explosion tends to produce vibration unless the engine and foundations are very secure.

THE DIESEL MOTOR.

Although not included in the scope of this paper I wish to make a brief reference to the new engine or motor that has been brought out in Germany and is, I understand, now being built in this country, the Diesel Motor. This is said to be the most perfect heat engine ever made, and one that in many tests has given from 25 to


Gallons
100,000.
200,000.
300,000.
400,000.
500,000.
Diagram No.1
Annual Cost
of Supplies
and Repairs
Gasoline and Oil Engines
Gasoline and Oil Engines
Compound Steam Engines
Average Daily Pumping
100.
200.
300.
400.
Diagram No.2.
Annual Cost
of Attendance
Gasoline Gas, Gas or Oil Engines
Compound Steam Engine
100.
200.
300.
400.
500.
600.
700.
800.
Gallons
100,000.
200,000.
300,000.
400,000.
500,000.
Diagram No.4
Annual Cost
of Fuel.
Gas @ 4.00 per 1000 cu. ft.
Oil @ 8c per gal.
Gasoline @ 16c per gal.
Steam Coal @ $3.00 per ton
Average Daily Pumping
1000.
2000.
3000.
4000.
500.
1000.

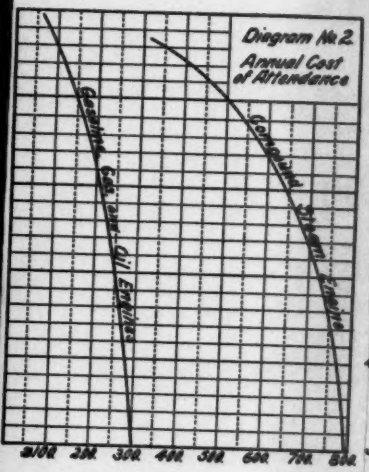
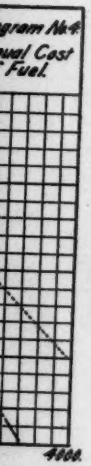
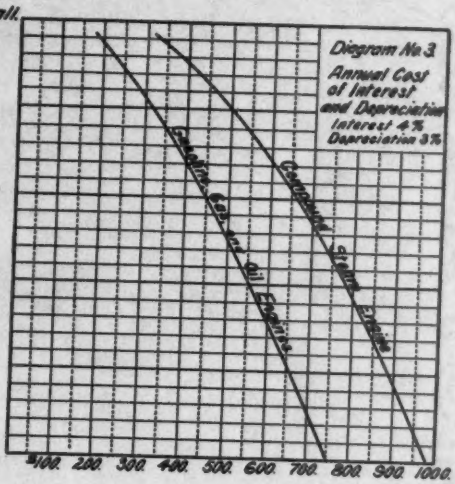
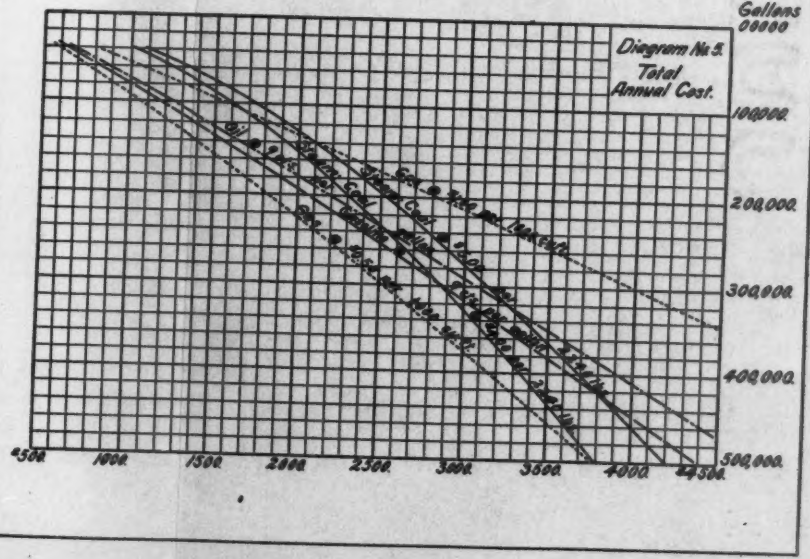


Diagram No. 2.
Annual Cost of Attendance
Gasoline Gas. and Oil Engines
Compound Steam Engine
100
200
300
400
500


gram No. 4
ual Cost
Fuel.
400


0000 gall.
Diagram No. 3.
Annual Cost of Interest and Depreciation
Interest 4%
Depreciation 3%
Gasoline Gas. and Oil Engines
Compound Steam Engine
100,000
200,000
300,000
400,000
500,000
100
200
300
400
500
600
700
800
900
1000


Gallons
00000
Diagram No. 5.
Total Annual Cost.
Average Daily Pumping
100,000
200,000
300,000
400,000
500,000
500
1000
1500
2000
2500
3000
3500
4000
4500

30 per cent. efficiency of the total heat energy in the fuel, as compared with 18 to 19 per cent. in the best gas engines, 16 to 17 in the best oil engines and 10 to 12 in the best steam engines. The fuel used is petroleum. If this engine has the advantages and economy that are claimed for it, it must have a notable influence upon the production of power.

A COMPARISON OF COMPOUND AND TRIPLE EXPANSION ENGINES FOR SMALL WATER WORKS PLANTS.

BY J. M. BETTON, ASSOCIATE MEMBER N. E. W. W. ASSN.

[*Presented Dec. 14, 1898.*]

The attention of water works superintendents and engineers has been drawn for the past few years to the question of economy in water works management. Every item of expense has been closely calculated and, by comparison with the work of others, made known through an interchange of experiences, they have reached astonishingly low results.

In no one direction has there been developed a greater opportunity for saving in the general operation of a pumping station than in the cutting down of the coal pile.

The problem of saving coal in cities and towns pumping a sufficient quantity to justify an expenditure for a high duty engine is comparatively easy of solution, but the desire and necessity of saving coal is no less in small towns where the consumption is light, while the proposition is much more difficult as the income will not justify an expenditure for refined machinery and the old pump must be kept at work, although it may be a coal eater and a bane to the life of the superintendent. In this unhappy condition the thoughtful manager of a small water works eagerly asks: "What can I do to economize in coal?" What pumping engine is there of small capacity, not beyond us in cost, and of fair economy that I can recommend to my board of commissioners as showing sufficient saving to justify the expenditure.

It is evident that the standard compound engine as made by various manufacturers of pumping machinery throughout the country does not meet this requirement.

He finds these engines of varying degrees of excellence, depending upon the care and skill with which they are built and erected, but of substantially the same design, as they are all modeled closely

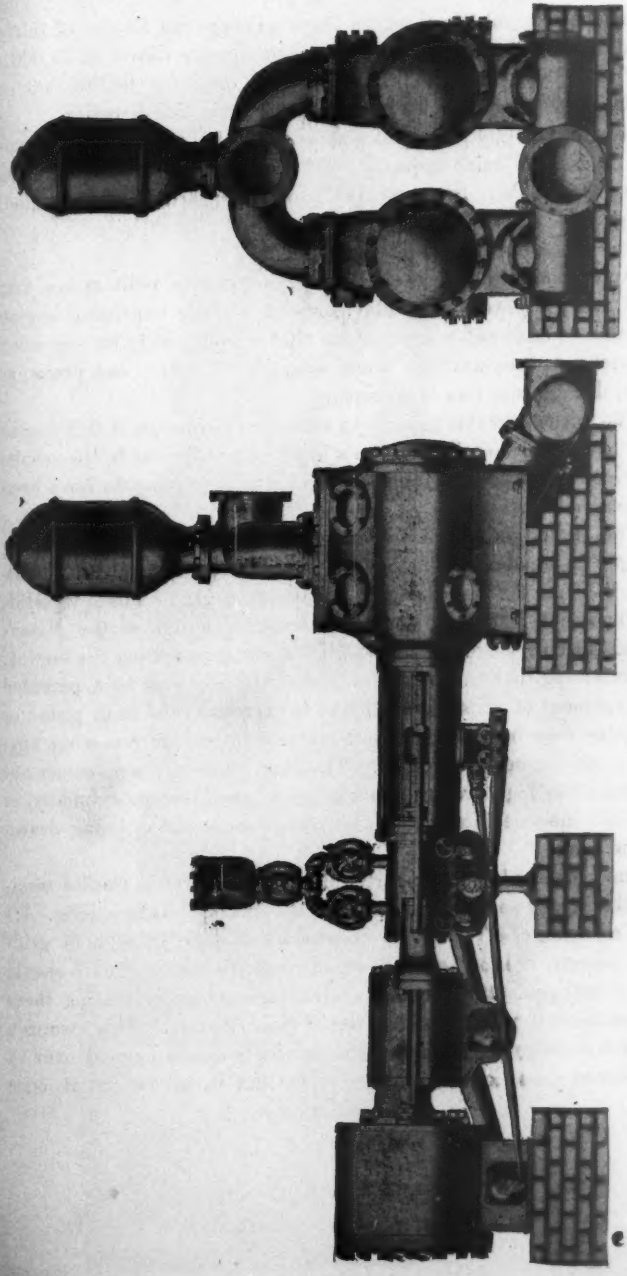


Fig. 1.

from the original Worthington Duplex Compound Engine of thirty years ago; and as he contemplates their running record of 25,000,000 or 30,000,000 duty, where he has hoped for 60,000,000 or 70,000,000, his case looks hopeless, at least in this direction.

It is to meet just this often expressed desire for a more economical engine of small capacity that Henry R. Worthington have brought out their Standard Horizontal Triple Expansion Engine. (Fig. 1.)

The problem before them was this:

An engine so simple in design that it could be built at low first cost and yet possess the refinements of a triple expansion engine capable of showing a coal saving that would justify its use when pumping a comparatively small quantity of water, and prove, as well, durable and free from repairs.

The purpose of this paper is to show you the design of this engine and to ask you to follow me in a brief calculation as to the results obtained by those now in use to see if the requirements have been successfully met.

The engine is of the 6-cylinder type having the cylinders arranged in tandem with the low pressure on the outside. (See Fig. 2.) At the first glance the objection might be raised that this arrangement made the engine inaccessible and that the pistons could not be removed without effort or else dismantling the engine. By referring to Fig. 3, however, you will see that by a patented arrangement of rods, the difficulty is overcome and each piston or cylinder may be inspected without dismantling or removing anything but the necessary head. The high pressure piston comes out between the high pressure and intermediate pressure cylinders, as does the intermediate piston, the low pressure piston being drawn at the back.

The steam valves (in nearly all cases) are of the Corliss semi-rotative type and are placed beneath the cylinders. (Fig. 4.) The advantages of this type of steam valve over the slide or grid-iron pattern are too well known to need discussion, but I should like to call your attention to the advantage gained by placing these valves beneath rather than on top of the cylinders. This insures a perfect drainage of the cylinders, as any moisture carried over by the steam does not enter the cylinder but is carried out at once

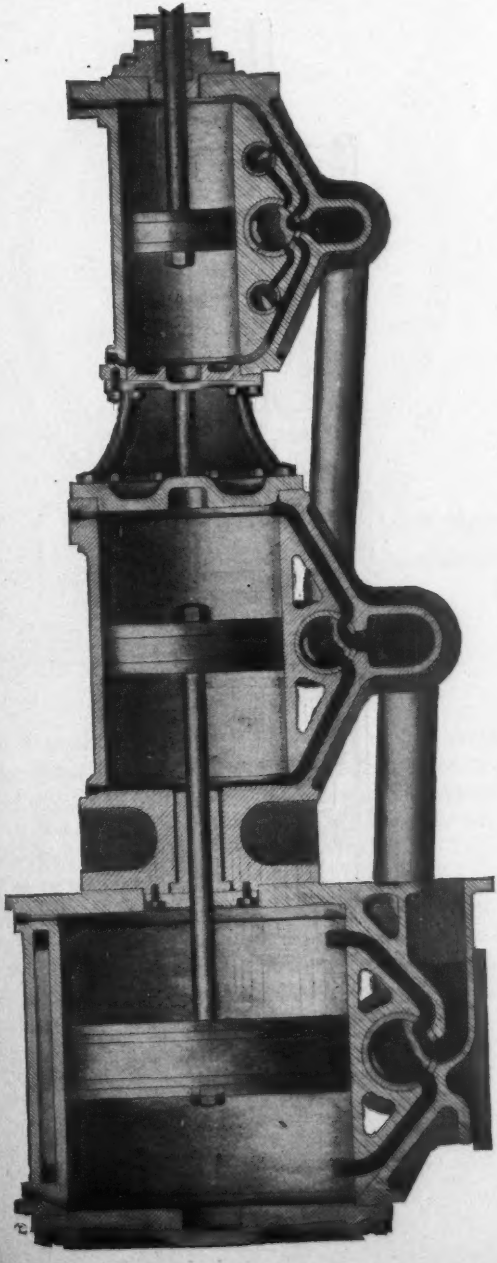


Fig. 2.

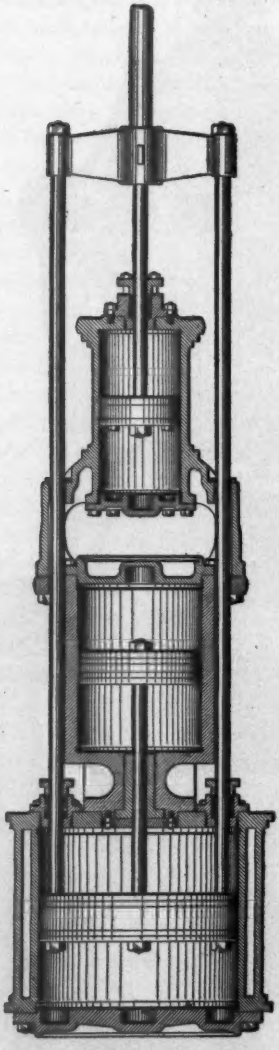


Fig. 3.

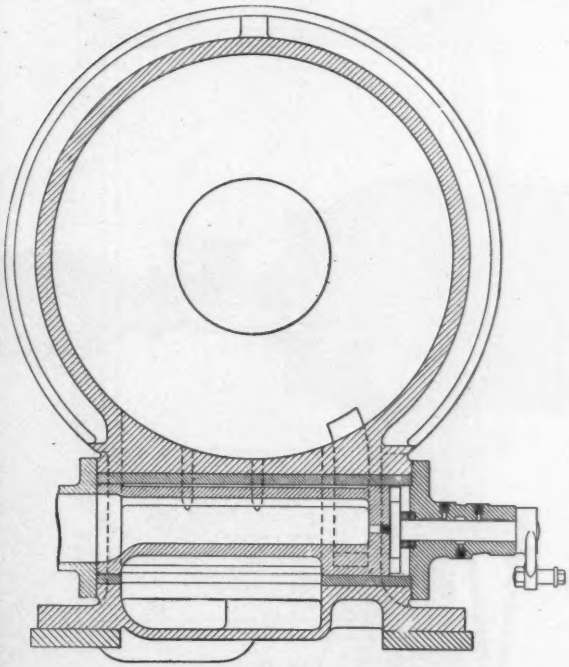


Fig. 4.

through the exhaust. In fact since adopting this arrangement, it has been found that it is entirely unnecessary to use drip valves or drains of any kinds on the steam cylinders. The cut-off valves are placed on the high pressure cylinder only and are set to cut off from $\frac{1}{2}$ to $\frac{3}{4}$ of the stroke.

Now as to economic results: It is a well known fact that the maximum duty obtained with the compound condensing direct acting engine (Fig. 6) is in the neighborhood of 50,000,000 to 70,000.000 foot pounds, depending upon its size, and an engine of this type that can steadily maintain such results is certainly making a good showing. With the triple, however, there is no difficulty in attaining and maintaining duties of from 70,000,000 to 100,000,000.

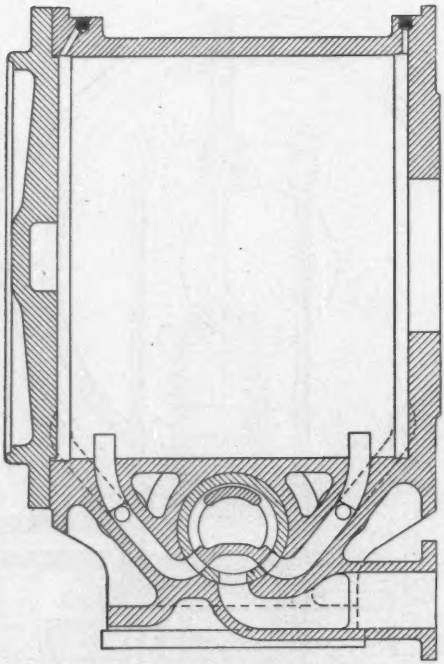


Fig. 5.

During the past year there have been installed upward of 20 of these engines of capacities ranging from 750,000 to 5,000,000 gallons per 24 hours when pumping against heads of from 20 to 1,000 feet.

At the Vulcan mine in Michigan the engine is delivering 1,200 gallons per minute against a pressure of 1,000 feet, and on the official trial showed a duty of 92,000,000 foot pounds per 1,000 pounds of dry steam. Subsequent tests of this engine made under every-day running conditions by the mine engineer have shown a duty of over 80,000,000 foot pounds, based on actual water evaporated in the boiler, no allowance being made for the condensation in 1,200 feet of steam pipe.

At Concord, N. H., the engine is of 2,000,000 gallons capacity,

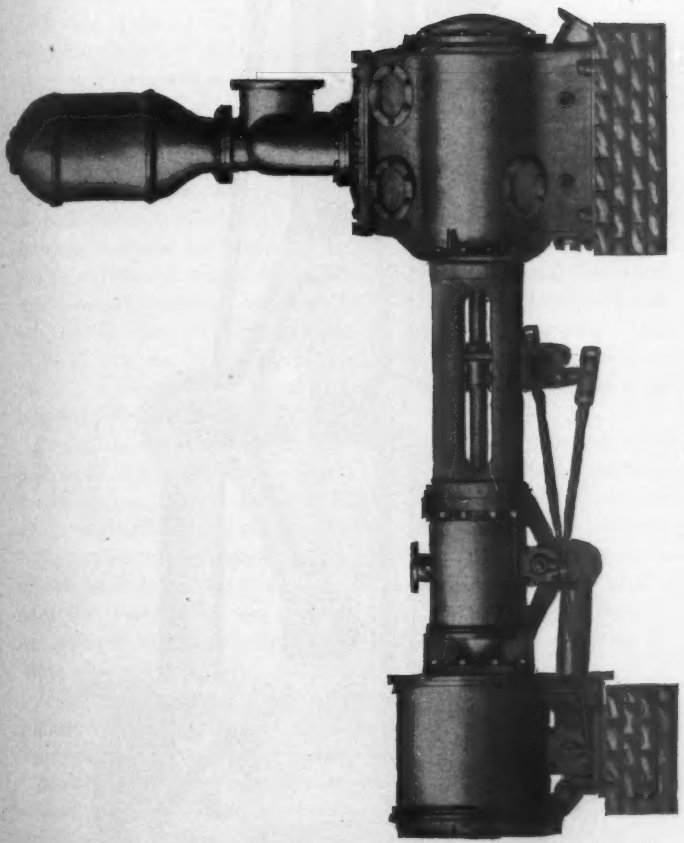


Fig. 6.

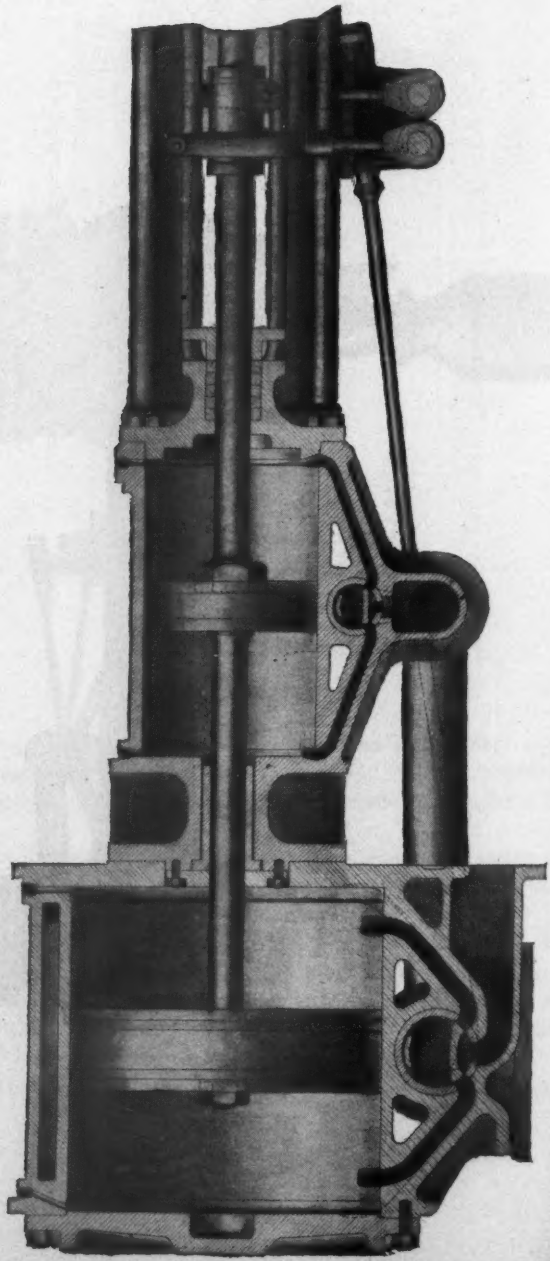


Fig. 7.

pumping against a pressure of about 80 pounds. The trial developed a duty of 82,000,000 foot pounds per 100 pounds of coal fired, no deduction being made for ashes, etc., and from a report received from the engineer, we learn that the average yearly duty on coal, including banking and heating and when running only an average of about four hours a day, is 73,000,000 foot pounds.

In the majority of cases the running engineers fail to keep accurate records of their engines, but it will be found that, in most every case, engines of this type are steadily maintaining the duties claimed for them.

Taking up the question of first cost for the purpose of comparison, let us take a compound condensing and triple expansion engine of say 2,000,000 gallons capacity in 24 hours working under 80 pounds head, all conditions being identical in both cases. The compound could be furnished complete with foundations and all piping within the engine house for about $3,600, the triple expansion with all the above for about $5,400. Estimating coal at $4 per ton, and the duty of the compound on the basis of an evaporation of 10 to 1, to be 55,000,000 and the triple 80,000,000, which we think is a conservative estimate and is giving the compound the benefit of the doubt, if any.

Fig. 7.

To run the compound for 24 hours would then consume 5,616 pounds of coal at a cost of about $11.20. The triple for the same length of time would consume 3,900 pounds of coal at a cost of $7.80, thus effecting a saving of $3.40 per day or $1,241 per year.

This saving is equivalent to 23 per cent. of the first cost of the engine, and deducting from it 6 per cent. on the original cost, or $324, for depreciation and ordinary wear, and 5 per cent., or $270, for interest, there remains a balance of $647 to the credit of the triple expansion engine.

If the net saving were capitalized at the rate of 5 per cent. per annum, it would represent $13,000, an amount amply justifying the first cost of the triple expansion engine.

Should the estimates be increased by the addition of boilers, it is evident that the triple expansion engine will require a boiler of smaller commercial rating than the compound condensing and at a correspondingly lower price.

The cost of operating the triple expansion engine, including

wages of engineer and fireman, oil and waste and the item of repairs, is no more than for a compound engine of the same capacity, and the comparison will be favorably maintained under different heads and capacities.

The above calculation of saving of the high class engine is based on the assumption that the plant is operated for 24 hours a day. When this is not the case the estimate must be modified to suit the existing conditions.

These are facts drawn from actual experience which are not generally appreciated and which it is my privilege to bring to your attention, in the hopes that, as your need of additional pumping machinery becomes apparent, you will call for the simplest and most economical type of engine, especially for moderate sized plants.

SUMMARY.

COMPARISON OF COMPOUND AND TRIPLE EXPANSION PUMPING ENGINES.

Capacity 2,000,000 gallons per diem. Head 80 lbs. Coal $4 per net ton.

	Compound.	Triple Expansion.
Cost	$3,600	$5,400
Duty (foot pounds)	55,000,000	80,000,000
H. P.	65	65
Coal per H. P.	3.6 lbs.	2.5 lbs.
Coal per day	5,616 lbs.	3,900 lbs.
Coal at $4 per ton	$11.20	$7.80
Coal per year	$4,088	$2,847
Saving per day		$3.40
Saving per year		$1,241
Per cent of cost	23 %	
Deduct for depreciation 6 % and 5 % for interest		$594
Balance in favor of triple expansion engine		$647

Equivalent to 11.8 % on original cost.

Capitalized at 5 % equivalent to $13,000.

THE LATEST DESIGNS IN WORTHINGTON PUMPING MACHINERY, COMPARING THEM WITH THE PRACTICE OF TWENTY YEARS AGO.

BY CHAS. C. WORTHINGTON.

[*Presented Dec. 14, 1898.*]

At the beginning of the period mentioned in the title of this paper, the Worthington water works pumping engine had been reduced for some years to practically standard lines of construction. It consisted of high and low pressure cylinders placed tandem, two on each side of the engine, which were provided with plain slide valves generally fitted with some form of balancing device. In this form it had been applied to upwards of 90 pumping stations, and had become very generally distributed throughout the United States. (Plate No. 1.) It disputed the field with a variety of crank and fly-wheel engines that were credited with a much higher duty performance, but which were of greater cost.

To the claims that have always been made for the Worthington engine of reliability, simplicity and uniformity of propulsion, it was necessary at this time to make the added one that these competitive engines could not save enough coal over that required by the Worthington to justify their extra cost. This commercial side of the competition brought out in the old days many discussions more or less heated, between the advocates of the two types of machines. Nor has the subject of the proper annual charge for investments in pumping machinery been by any means exhausted since. It is rather strange that this calculation, that involves the most ordinary principles of investment, is so little understood by many writers on the general subject of water works installations. It is not unusual to see the extra purchase price of an engine charged with only 5 per cent. interest in making comparisons between costs. In these days, when engineers are taking very practical and business-like views of all matters pertaining to the economic administration of

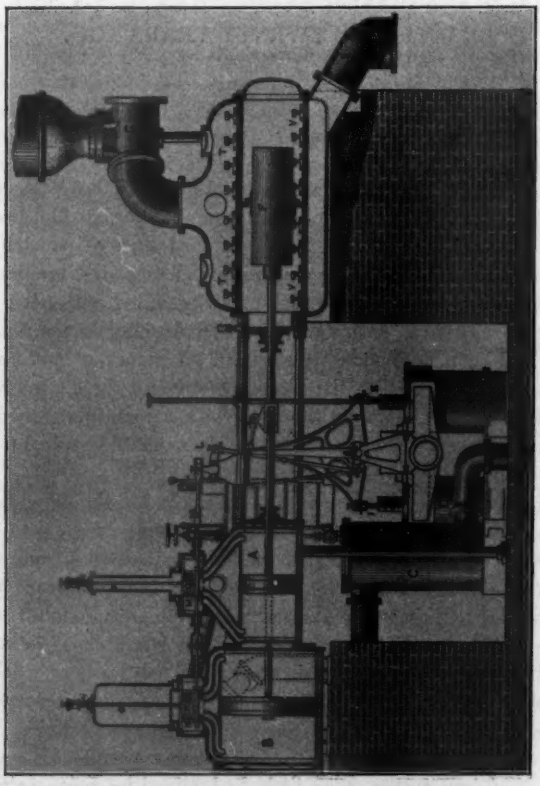


Plate No. 1.

SECTIONAL VIEW OF STANDARD FORM OF WORTHINGTON ENGINE IN 1880.

their work, and when they are relied upon to counsel the investor as to the ultimate returns to be gained by a given expenditure, it is of the utmost importance that this question should be definitely settled and the same rule of calculation adopted by all alike. If a proper percentage were charged against some of the recent investments that have been made for high class pumping machinery, it would often be found that those engines, which take first rank in economy of fuel, were really the most costly that the city could purchase.

It is plain to see that in arriving at this proper charge, the following items should be considered:

First. Simple interest upon the investment. This, for the purpose of discussion, may be taken at 3 per cent.

Second. A fair percentage, to cover the items of repairs and maintenance. No experienced engine builder would agree to bear the expense of the up-keep of his engine for less than an annual charge of 3 per cent.

Third. An additional percentage, to form a sinking fund, which shall equal the original expenditure when the engine becomes worn out, or is replaced by a more up-to-date machine.

In view of the advance during the last few years in the art of engine construction, and the tendency of the times towards radical changes in old methods, it is safe to predict that the majority of the best engines built today will be replaced, as obsolete and wasteful, twenty years from now. This is a fair estimate of the lifetime of many of our modern engines. But even extending this period to twenty-five years, the sinking fund imposes an annual charge of 4 per cent. This, then, taxes our original expenditure with an interest rate of 11 per cent. Taking this at 10, to escape all cavil, it still remains a sum much higher than is generally selected by writers on the subject.

In the early discussions to which we have referred, not only was this interest account of vital importance to the claims of the Worthington engine, but also the question of what was the actual amount of coal an engine giving a higher test duty could practically save. Often the comparison was distorted by estimating that the calculated saving per horse power per hour would be secured for the 24 hours of the day, 365 days in the year. You gentlemen, who

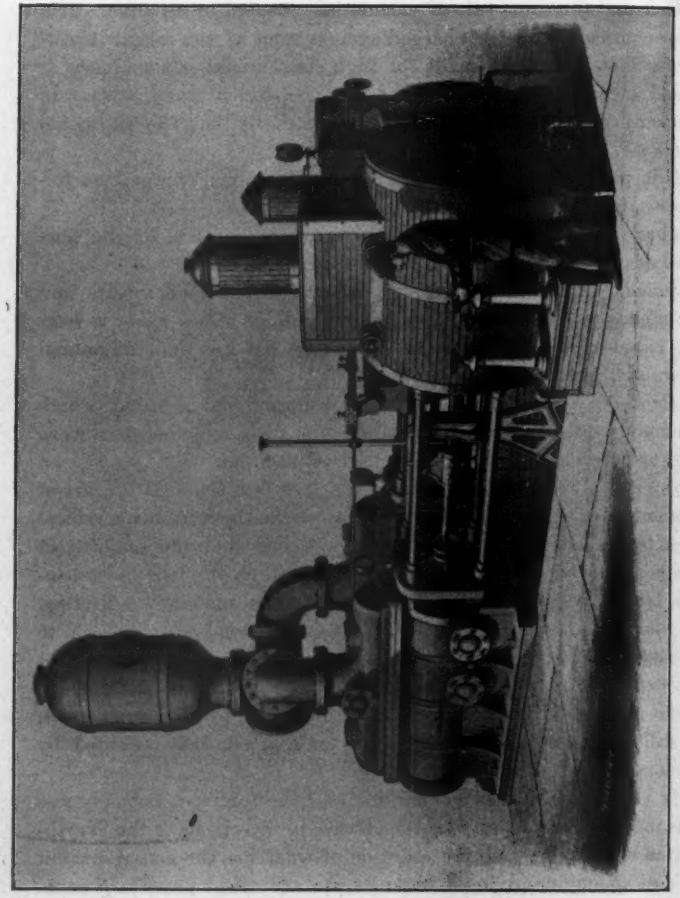


Plate No. 2.
WORTHINGTON TAKE ENGINE, 1885.

have personal experience in the running of pumping stations, appreciate that few pumping engines are ever called upon to work continuously during the whole year. In making the comparison between engines of different grades, it is as true today as it was twenty years ago, that the number of hours in which the engines are to be run should be taken into account, as they represent the only time in which the estimated saving of the higher class engine can be secured.

This commercial argument was for a number of years strongly on the side of the Worthington engine, because the difference between its cost and that of the engines with which it was in competition was very considerable. But as less expensive designs were adopted, and as the duties were gradually improved, this difference became less marked, and the calculation less favorable. Finally, the builders of the Worthington engine were forced to the conclusion that something would have to be done to raise its economy, or it would be driven out of the field of high class water works stations altogether. It was also sought to decrease its cost. Experiments were conducted with what was called the tank engine (Plate No. 2), whose steam end consisted of two cylinders, placed side by side, instead of four of the older type. It was thought that by decreasing the number of cylinders the cost would be reduced. It was found, however, that the practical difference was inconsiderable, while the performance of the engine itself was less satisfactory. The ratio between the high and low cylinders of the 4-cylinder type was usually fixed at about $3\frac{3}{10}$ to 1, while in the tank pattern it was made from $7\frac{1}{2}$ to 8 to 1. In point of economy, there was little difference between the two types, but the tank engine was more difficult to adjust. It was found that the balance of the two sides of the engine was thrown out somewhat by the variations in steam or water pressure, commonly met with in service. In view of the recent able and instructive presentation by Mr. Rockwood, of the theory of his engine, the performance of this one, with its high ratios of cylinders, takes on a reflected interest.

During these experiments with the tank type, it was suggested to place two complete pumps side by side, and exhaust the one into the other, in the way that has since been done in a feed apparatus for marine boilers, recently made the subject of a paper before the

Plate No. 2.
Worthington Tank Engine, 1886.

American Society of Mechanical Engineers. The action of the two machines, and the effect upon the water column, were so irregular, however, that it was deemed an impracticable method of securing the benefit of compound cylinders in engines of large size. The speed at which one engine ran with respect to the other was found to vary with the fluctuations of the water pressure.

At the time of which we are writing, the best duty to be secured by either the tank or 4-cylinder type of the Worthington engine might be placed at 29 pounds per horse power. To raise this, then, became the object sought for by its builders. The engine had gained for itself an enviable position; its great simplicity and adaptability to all kinds of service were universally admitted; and it had been reproduced on the same lines of construction more frequently than any other pumping engine of that day or now. It may be said to have been the first pumping engine whose design had been reduced to standard patterns, and its low cost, the cheapness of its foundations and of the building required to accommodate it, gave it a strong hold on the less pretentious stations of the country, from which even now, with all the competition of high duty types, it is difficult to dislodge it.

Little could be expected in the way of improved efficiency by any change in the arrangement or ratios of the cylinders themselves. The necessity of maintaining a uniform speed of propulsion throughout the length of the stroke prevented any greater proportion of cylinders being used than about 4 to 1, and yet it was realized that unless the number of expansions could be increased, there was little chance of any marked improvement in the steam consumption of the engine. It was at times feared that perhaps the duplex engine had reached its ultimate results, and that it was destined to be relegated to a position second-rate among the high class water works engines of the world. Many of its warmest adherents suggested that the fly-wheel offered the only chance for improvement, and the Holly-Gaskill engine, which at that time was making inroads into the field, and which embodied, apart from its fly-wheel, many features of construction similar to those of the Worthington engine, was pointed at as a significant warning.

Being unwilling to sacrifice the traditions of the business or to abandon the admitted advantages in handling water which the

duplex motion possessed over any motion derived from the action of a crank, we began a search for some equivalent to the fly-wheel. Devices embodying the principle of tumble-bobs where a weight would be lifted at the first part of the stroke and in descending would give off its reserve power at the latter part, as well as numerous others, embodying the principle of momentum, were investigated, but all were considered impracticable from one cause or another, but in the main because they added the danger of greatly increased weight of moving parts to the inherent difficulty of stopping the engine at the end of the stroke. What was necessary was a power that, while sufficient in itself to represent the work of a fly-wheel, of any size whatever, would leave the engine practically as free to start and stop as if it were not exerted. Such an attachment was finally found, and converted the engine into what is now known as the Worthington High Duty. This attachment has been so often described and so many of the engines have been built, that a detailed description of it here is not necessary. Suffice it to say in brief that the principle embodied in the compensating cylinders is capable of producing the same effect upon the steam diagrams of the engine as a fly-wheel could produce. In other words, there is nothing inherent in their application which prevents the steam being utilized in the engine to the highest point of economy at present obtainable in engineering practice.

At the start, some difficulties in construction naturally were met with. The experiment was along unknown lines. No books had been written on the subject, no experience had been gained which could serve as an aid in carrying out the improvement. This, we think, should not be lost sight of in discussing today the relative theoretical merits of the direct acting and the fly-wheel engine. While the one embodies the accumulated experience and effort of over a century of the world's best engineering talent, and has been brought to its present state through the crucible of many mistakes and failures, all of which have served as guides and danger signals to the present constructors, the other stands as the result of the unaided effort of one company to produce an engine that had to be a success from the start, from an engineering standpoint not only, but from a commercial one as well. When the time comes for the minds of engineers at large to be directed towards the principles in-

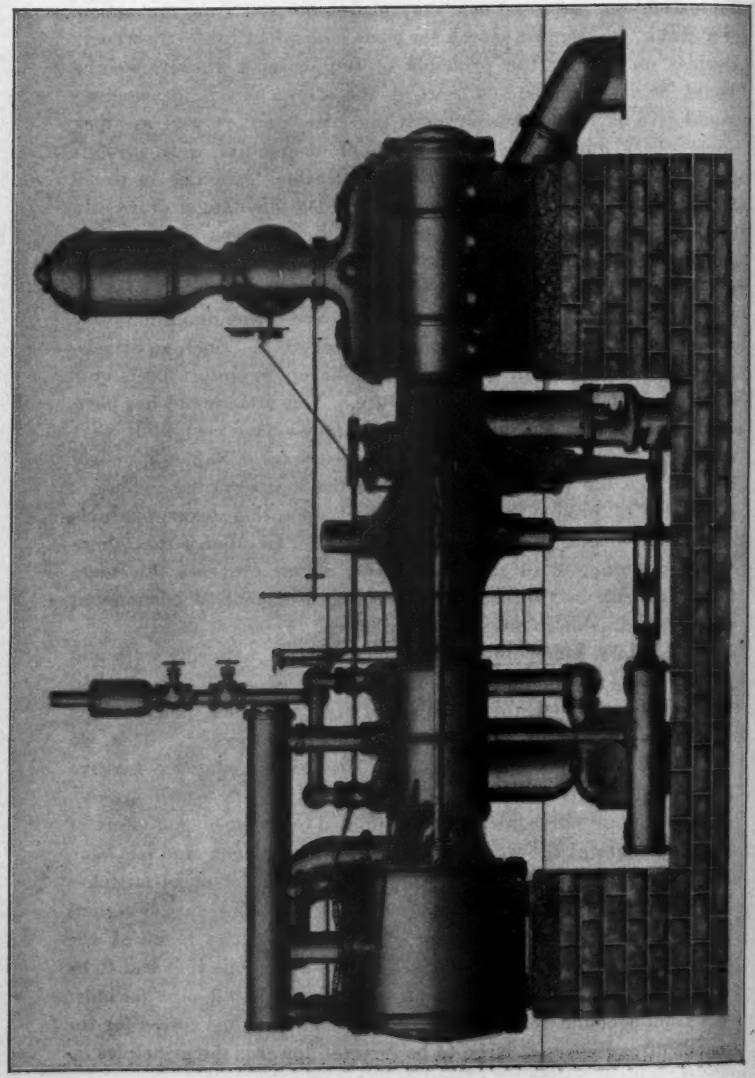
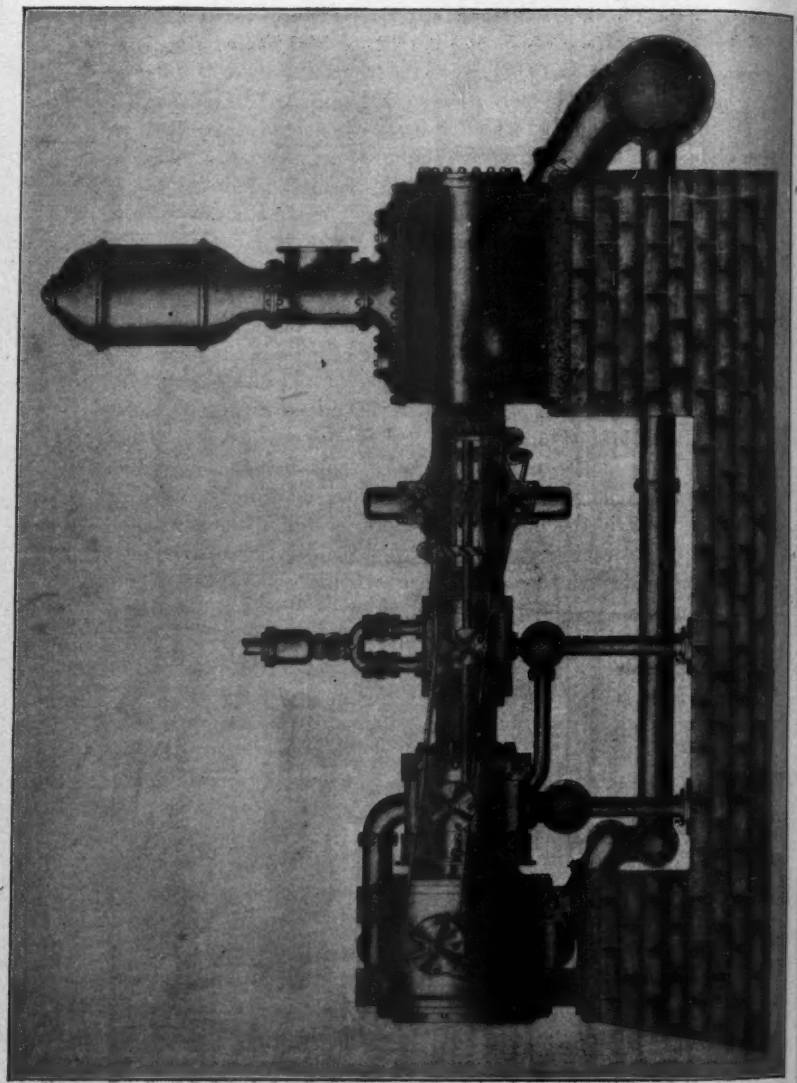


Plate No. 8.
Worthington Compound Condensing High Duty Pumping Engine, 1895.

volved in the Worthington High Duty Engine, we believe that those principles will be found enduring and that in their hands, when ours perhaps have "knocked off" work, the future history of the engine will be far brighter and more prominent than that of its older rival.

On the first engines to which this attachment was connected, as few changes in the former construction of the machine were made as circumstances permitted. Practically the only addition to the old engine, other than the attachment itself, was the cut-off valves. The old form of slide valve was adhered to. The compound condensing engine, thus fitted with compensators, at once achieved a duty of about 19 pounds per horse power. From that time on, such improvements in its details as have been made have enabled the same engine to show 16 pounds. The changes in the steam cylinders related principally to the location of the valves. It was found that the slide valve could not be relied upon to be tight under the new conditions of higher steam and greater expansion and that its steam ports contained more clearance space than was desirable. This led to a change from the slide to the semi-rotative type of valve. These, in the first engines, were located at the top of the cylinder, one at each end, and performed the function of both inlet and exhaust valves, the cut-off valves being separate ones and placed directly over them. (Plate No. 3.) While this reduced the clearance of the engines, simplified its valve motion and improved its economy, it was thought that still better results could be secured if the inlet valves could be made to perform the function of cut-off as well, and could be separated from the exhaust ones with an arrangement of valve chests, such as are common to the Corliss engine. This required, however, a valve motion of special design, as the ordinary trip motion of the Corliss gear is inapplicable to an engine of the direct acting type. Within the past two years the valve motion illustrated in the accompanying Plate No. 4, has been adopted, and in its principle of operation is believed to be unique. As will be seen, the admission valves, which are also the cut-off ones, are located at the top of the cylinder and the exhaust valves at the bottom. These latter are operated by direct connection with the disc, which disc is vibrated back and forth by link connections to the rockshaft operated by the steam cylinders on the opposite side. When this disc vibrates, that which may be described as the

Plate No. 3.
Worthington Compound Condensing High Duty Pumping Engine, 1895.



"four-armed crank," which is pivoted to the disc, is made to move with it in such a way as to open the valve on one end of the cylinder for the admission of the steam. While making this movement it is fulcrumed on the point of connection with the link that is driven by its own engine. When steam is thus admitted, and the engine starts on its stroke, the disc being then stationary, the four-armed crank pivots on the point of connection with the disc, and is moved by the other link in a reverse direction, thus cutting off the steam. Through the angles of the cranks on the cut-off valves, this cut-off motion is remarkably rapid, and produces a steam diagram that is all that can be desired. Any point of cut-off can be secured by changing the location in the four-armed crank of the sliding block which carries the valve link. It will be seen from this that there is no trip or lost motion to this gear. It is absolutely noiseless in its operation, and as the work which it has to perform is done through long leverages, the wear on the pins and bearings is reduced to a minimum.

Following the demand for still higher duties, Triple Expansion Worthington Engines have been introduced during the last few years. These, particularly when used in connection with the higher steam pressures, that are at present met with, can be relied upon to secure a duty of 14 pounds per horse power. Usually, the steam is cut off in all three of the cylinders, so as to do away with any "drop," or expansion, without work. We are inclined to agree with Mr. Rockwood, that the precaution which we, in common with some other engine builders, take to prevent this drop, may be unnecessary. In the absence of any conclusive experiments in this direction, we have simply followed the general practice on theoretical lines. In this connection it is to be hoped that before long an engine for experimental purposes will be built, through the efforts, perhaps, of an Association like your own, of such size and power as to make any results reached in experimenting with it conclusive, and by which questions like this one, that are constantly being asked, could be authoritatively answered. The one that Mr. Rockwood has raised is of enough importance in itself to justify the expense of such an undertaking. Coupled with this could be the equally important one of the effect of high speed on the economy of an engine. While it is the custom to assert that there

is a saving in fuel to be secured by an immoderately high speed engine over a moderately slow one, the fact is that there is nothing positive known about it at all. A statement that a high speed engine is, *per se*, more economical than a slow speed one, is pure guess work from any data that we have today. No one can speak with assurance, either, as to the economy of jackets; no one knows what the exact effect is of superheated steam, nor the temperature at which this effect can best be utilized.

These and perhaps other equally important questions are constantly arising, many of which could be set at rest by a series of trials conducted by a board of experts on an experimental engine of the kind herein suggested. The time has come, we think, when a concerted effort in this direction should be made by the societies of the country, among which none could be more interested in the results than yours.

Perhaps the latest departure from the old method of construction of the Worthington, has grown out of the necessity of meeting the demand for vertical engines. Numerous Worthington engines of this type, both of the compound and triple expansion class, have been erected, an example of one being shown in Plate No. 5. In general principles it does not vary from the horizontal pattern, but the necessity of taking care of the weight of the moving parts of the engine, when in a vertical position, requires the addition of a balancing device. In its usual form this consists of a plunger of such area that, when acted upon by air pressure enclosed in a tank, it will exert an upward lifting force exactly equal to the superimposed weight of the rods and pistons. This tank is of such volume as not to materially change its pressure under the alternate compression and withdrawal of the air under the action of the plunger. As the tank is tight, and the leakages around the plunger are protected by water, no practical loss of this air pressure is sustained.

Since the date of the invention of the high duty attachment, 1886, well within the period we are considering, there have been built for water works stations more than 1,400 Worthington engines. About 15 per cent. of these in number, and by far a larger proportion measured by horse power, have been of the high duty type. This record, which would indicate that more Worthington engines

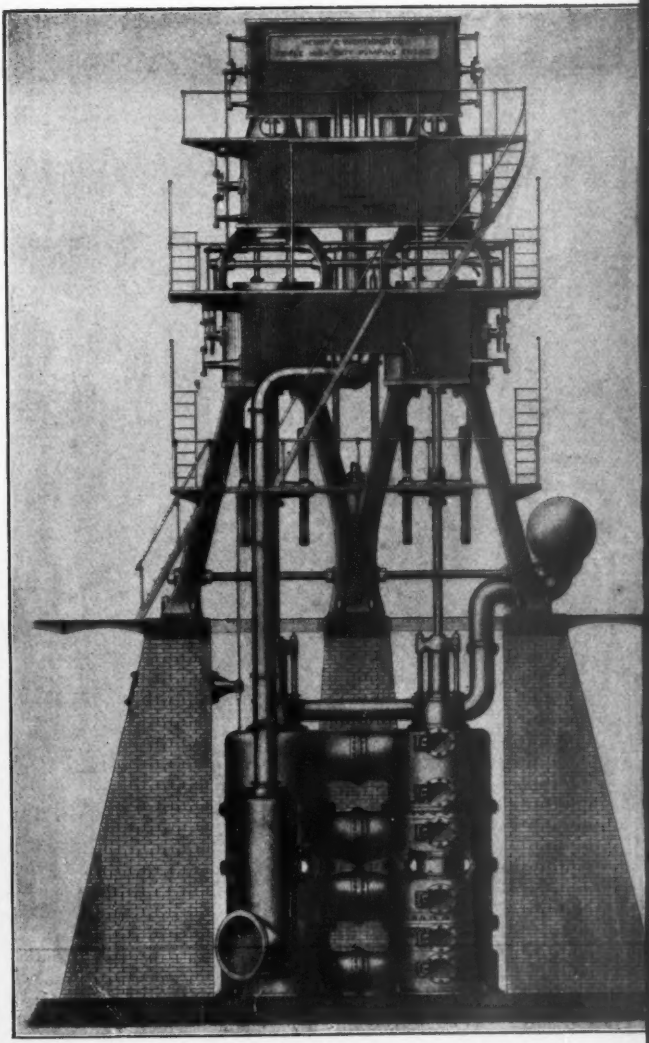


Plate N

WORTHINGTON TRIPLE EXPANSION VERTICA

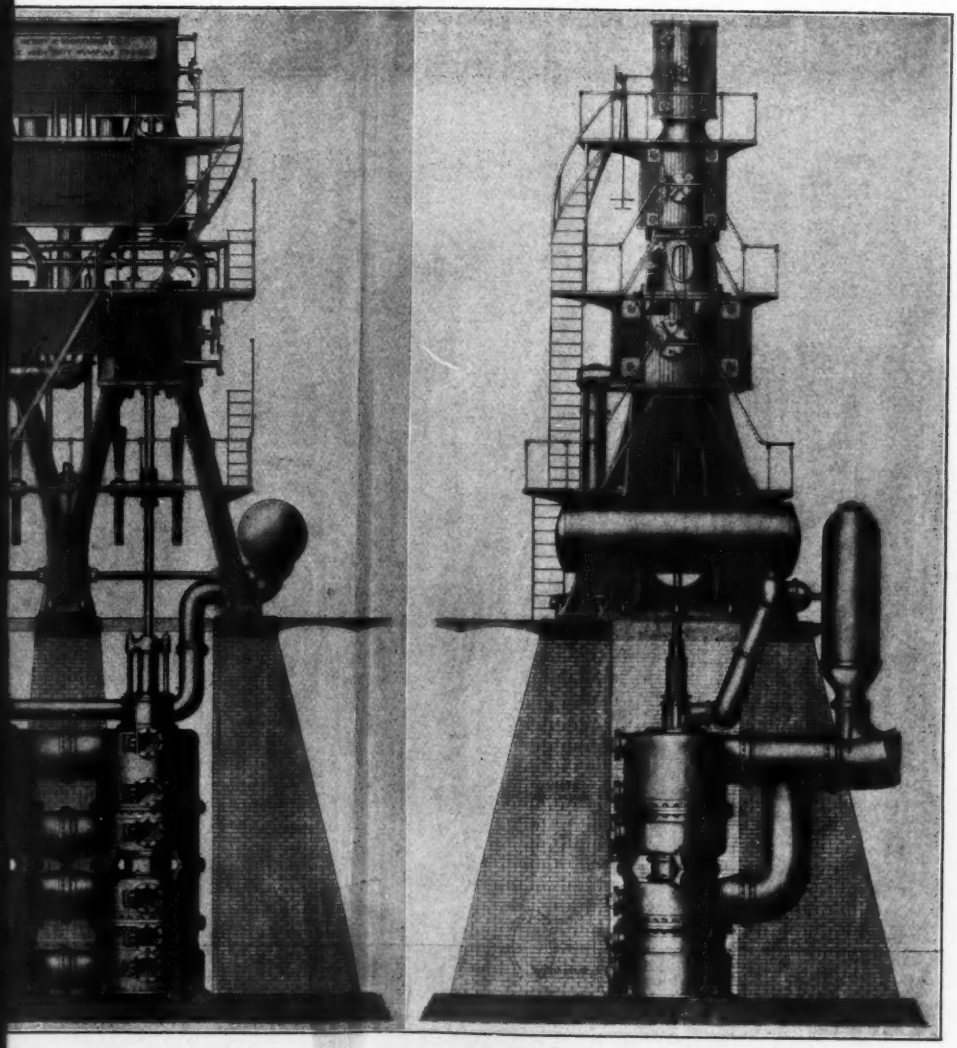


Plate No. 5.

WORTHINGTON TRIPLE EXPANSION VERTICAL HIGH DUTY PUMPING ENGINE, 1898.

have been constructed during that period than of all other types of pumping engines combined, could never have been approached were it not for the discovery of the means that we have briefly described, whereby the economic efficiency of the duplex engine was more than doubled without eliminating those peculiar merits upon which its old reputation was based. Perhaps never before in the history of engineering has there been an instance where so important and well known a machine, built on approved and standard lines, has been so markedly raised in efficiency in so short a time, and with so few changes in its principles of operation and details of construction.

PROCEEDINGS.

QUARTERLY MEETING.

YOUNG'S HOTEL,

Boston, December, 14, 1898.

President Forbes was in the chair, and the following members and guests were present:

ACTIVE MEMBERS.

Charles H. Baldwin, Lewis M. Bancroft, George E. Batchelder, Joseph E. Beals, James F. Bigelow, Dexter Brackett, E. C. Brooks, George F. Chace, Charles E. Chandler, Benjamin R. Chapman, John C. Chase, William F. Codd, Freeman C. Coffin, R. C. P. Coggeshall, Byron I. Cook, Henry A. Cook, Francis W. Dean, Charles R. Felton, Richard J. Flinn, F. F. Forbes, Frank Baldwin French, Frank L. Fuller, Julius C. Gilbert, D. H. Gilderson, Albert S. Glover, W. J. Goldthwait, Frederick W. Gow. E. H. Gowing, John C. Haskell, Louis E. Hawes, T. G. Hazard, Jr., Horace G. Holden, Willard Kent, Frank C. Kimball, George A. Kimball, James W. Locke, Arthur D. Marble, A. E. Martin, Frank E. Merrill, Leonard Metcalf, Thomas Naylor, Frank L. Northrop, W. W. Robertson, Henry W. Rogers, W. J. Sando, John E. Smith, J. Waldo Smith, George A. Stacy, Frederic P. Stearns, Lucian A. Taylor, Robert J. Thomas, William H. Thomas, D. N. Tower, William W. Wade, J. Alfred Welch, John C. Whitney, George E. Winslow.

ASSOCIATE MEMBERS.

The Edward P. Allis Co., by Irving W. Reynolds and Arthur West.
The George F. Blake Manufacturing Company, by George J. Foran.
Coffin Valve Company, by J. Alfred Welch.
Crosby Steam Gauge and Valve Company, by Robert Pirie.
Deane Steam Pump Company, by C. L. Newcomb.
Hersey Manufacturing Company, by James A. Tilden.
Ludlow Valve Manufacturing Company, by H. F. Gould.
McNeal Pipe and Foundry Company, by J. M. Holmes.
Mueller Manufacturing Company, by M. G. Millikin.

National Meter Company, by Mr. Lufkin.
Neptune Meter Company, by Mr. H. H. Kinsey.
Perrin, Seamans & Company, by H. L. Bond.
Rensselaer Manufacturing Company, by F. S. Bates.
Builders' Iron Foundry, by T. C. Clifford.
A. P. Smith Manufacturing Company, by W. H. Van Winkle.
Union Water Meter Company, by F. L. Northrop.

HONORARY MEMBERS.

G. H. Partridge of the "Engineering Record."

GUESTS.

A. H. Keene, C. E. Riley, W. B. Webber, A. H. French, H. T. Gibbs, J. G. Moore, Mr. Rockwood, Julian P. Wood, Harry L. Thomas.

The following named persons were elected members by ballot cast by the Secretary:

RESIDENT ACTIVE.

Frank Livermore Pierce, of Newton, Special Inspector with the Factory Mutual Fire Insurance Company.
Harry L. Thomas, Assistant Superintendent, Hingham Water Company, Hingham, Mass.
John N. Ferguson, Boston, with Metropolitan Water Works.
Henry R. Johnson, Water Commissioner, Reading, Mass.
George J. Foran, with the George F. Blake Manufacturing Company.

The President read a short paper on "Pumping Engines," which served as an introduction to the other papers, all of which related to the same general subject.

George H. Barrus, M. E., of Boston, contributed a paper entitled, "Possibilities of Economy in Pumping Engines, as Based on the Latest Accomplishments." The paper was read by Mr. Gowing.

The next paper was prepared by Mr. Reynolds, a representative of the Edward P. Allis Company of Milwaukee, Wis., and was read by Mr. Chace. The subject was, "The Latest Designs in Pumping Machinery, Comparing Same with Practice of Twenty Years Ago."

Mr. Freeman C. Coffin, C. E., of Boston, read a paper entitled, "The Application of Oil and Gasoline Engines to Pumping Machinery."

John E. Smith, Superintendent, Andover, Mass., presented a

"Comparison Between Low and High Duty Pumping Engines on a Small Water Works Plant."

Mr. J. M. Betton, M. E., of Brooklyn, N. Y., was expected to have been present and to have read two papers, but he was prevented by business engagements from attending, and his papers were not read. The subjects were, "The Advantages of Triple Expansion Pumping Engines," and "The Latest Designs in Pumping Machinery, Comparing Same with Practice of Twenty Years Ago," the latter paper being prepared by him as the representative of Henry R. Worthington of Brooklyn.

There was oral discussion by Mr. F. W. Deane, Mr. Rockwood, Mr. Reynolds and Mr. Smith.

Adjourned.

Table of Contents.

		Page.
Paper.	"Stave Pipe—Its Economic Design and Use," by Arthur L. Adams, C. E.	247
	Discussion by Messrs. Walker and Richards.	287
Paper.	"Improved Wyckoff Water Pipe," by George L. Wells, C. E.	288
Paper.	"Salt Water Fire System of Boston," by Frank A. McInnes	304
Paper.	"The Use of Steel for Water Mains,' by L. M. Hastings, C. E.	314
Paper.	"Water Pipes on Metropolitan Water Works," by Dexter Brackett, C. E.	325
	Discussion by Messrs. Porter and Fitzgerald.	331
Paper.	"Cleaning a Water Main in St. John, N. B.," by William Murdoch, C. E.	333
	Discussion by Mr. Brackett.	341
Adjourned Meeting.		345
Attendance		345
Election of New Members		346
Obituary—David B. Kempton, Geo. E. Batchelder, Jesse Garrett		349

48-in. Pipe Line—San Gabriel Power Co.